VISIT…

SECTION 15

BALLS

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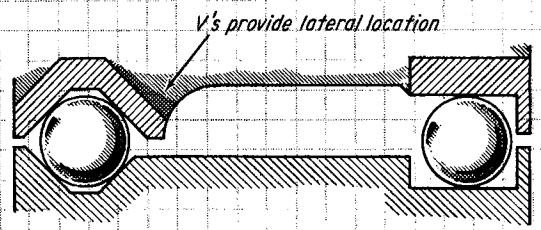
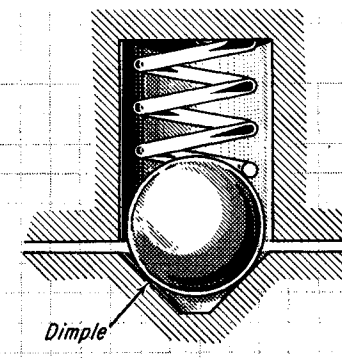
12 Ways to put Balls to Work

Bearings, detents, valves, axial movements, clamps and other devices can all have a ball as their key element.

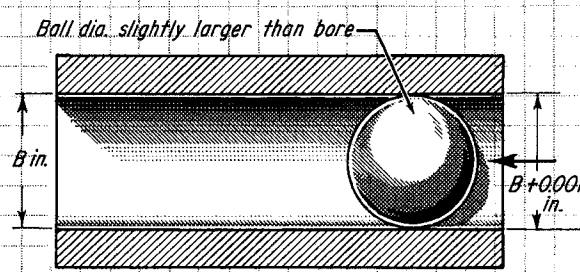
Louis Dodge



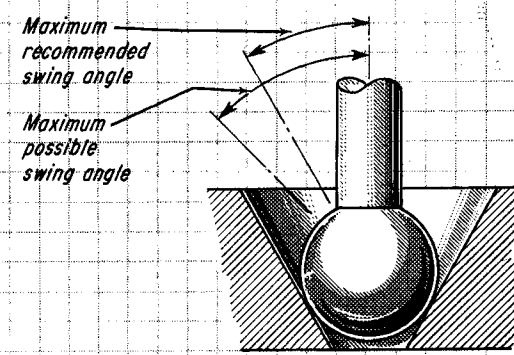
1 BALL-BEARING MACHINE WAY HAS LOW FRICTION.



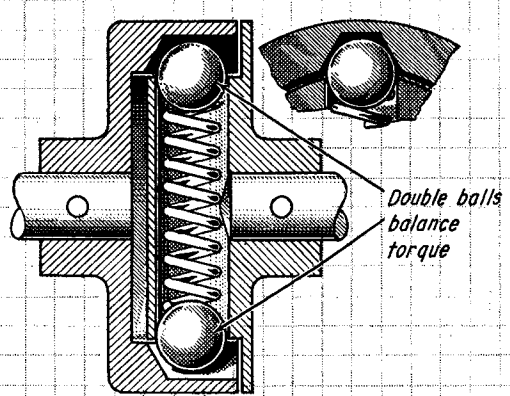
2 DETENT POWER DEPENDS ON SPRING STRENGTH AND DIMPLE DEPTH.



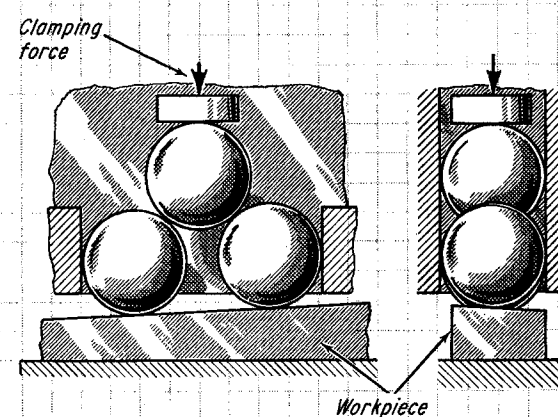
5 BALL ACCURATELY FINISHES BUSHING BORE.



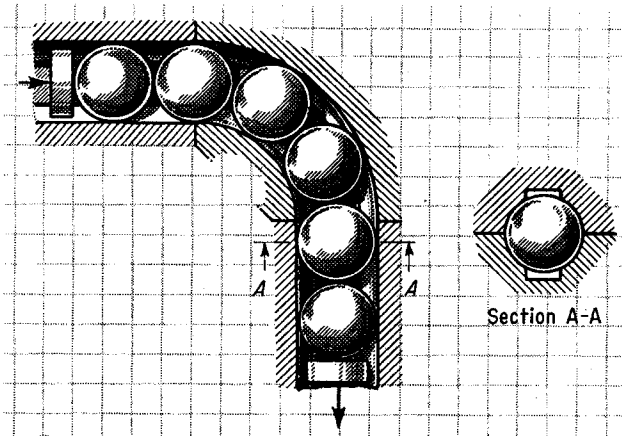
6 BALL SHAFT-END LETS SHAFT SWING.



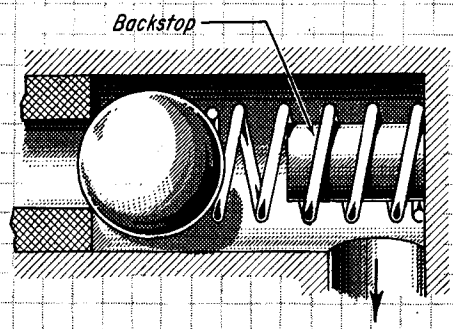
9 CLUTCH HAS LIMITED TORQUE TRANSMISSION.



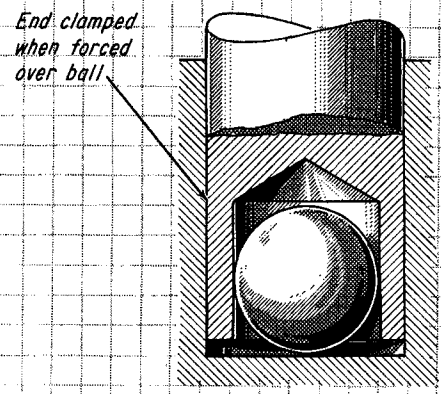
10 CLAMP UNEVEN WORKPIECES.



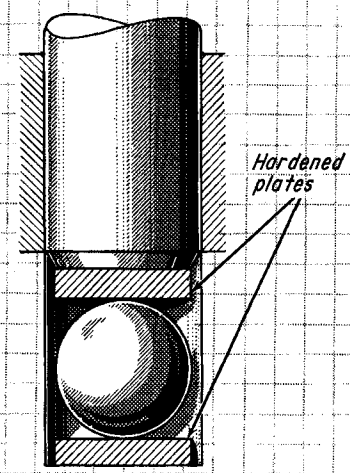
3 TRANSMIT AXIAL FORCE AROUND CURVES.



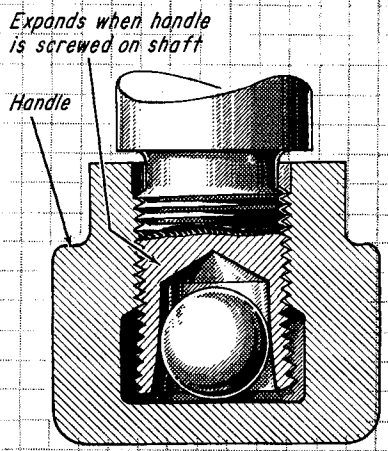
4 CHECK VALVE BACKSTOP IS ADVISABLE.



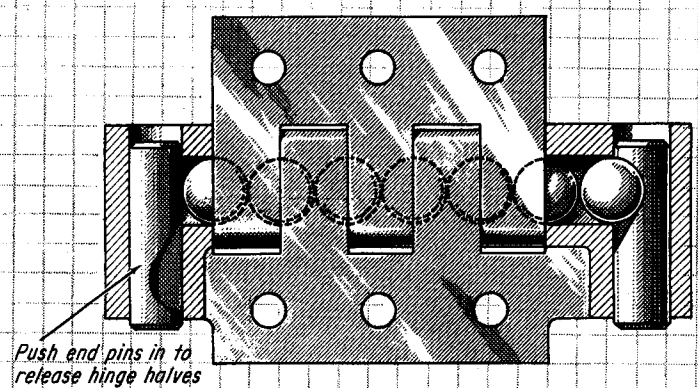
7 BALL-LOCK FASTENS STUD IN BLIND HOLE.



8 THRUST-BEARING TAKES LIGHT LOADS.



11 BALL-LOCK SECURES HANDLE TO SHAFT.

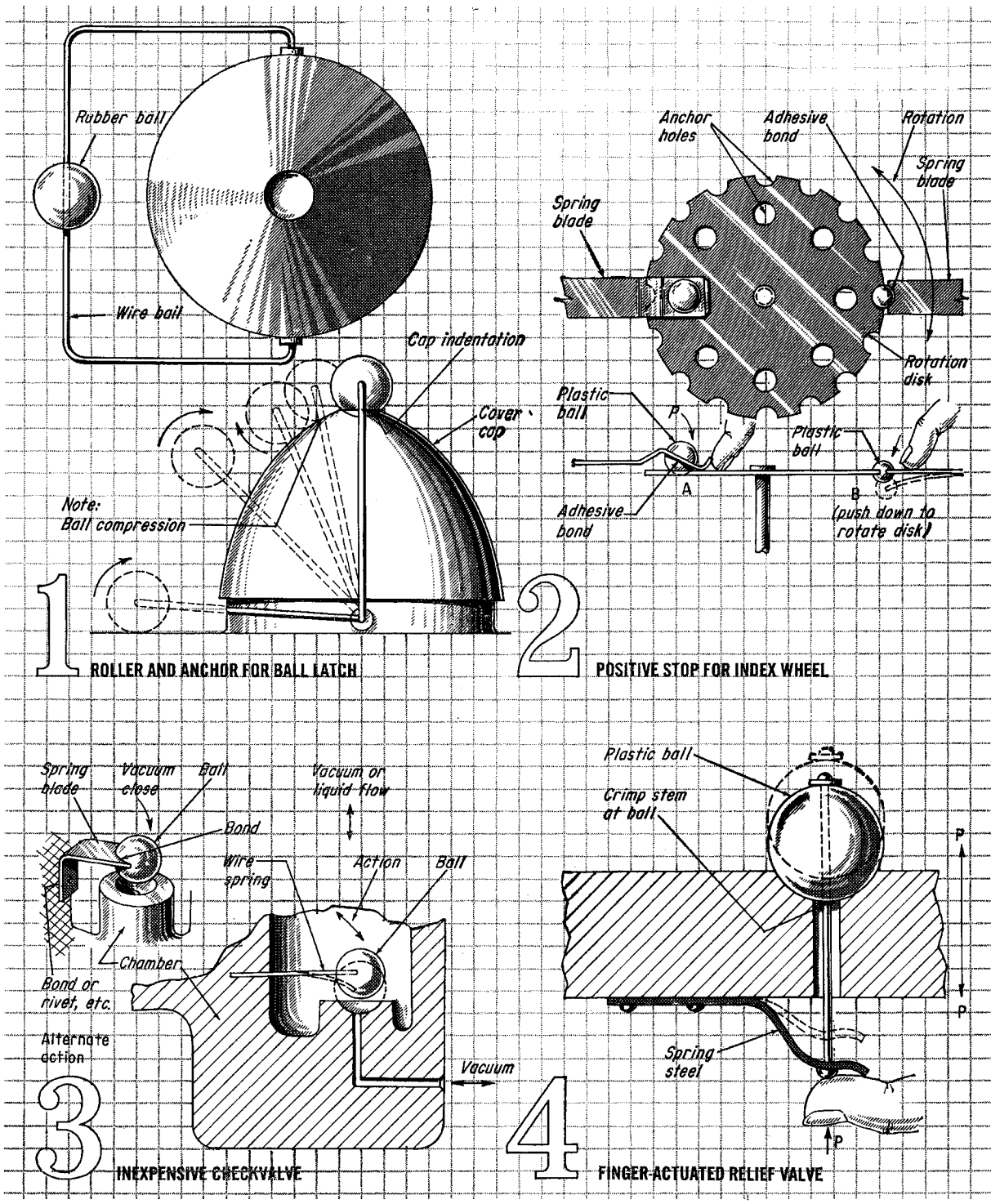


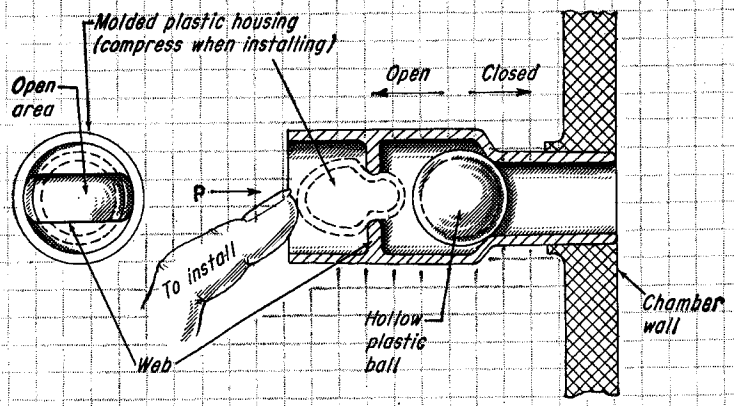
12 HINGE PIN IS SEPARABLE.

How Soft Balls Can Simplify Design

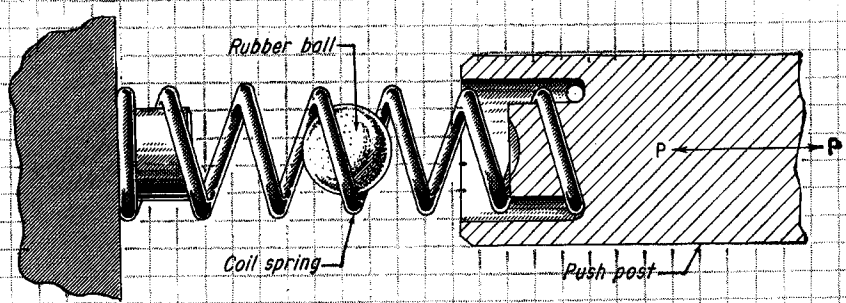
Balls of flexible material can perform as latches, stops for index discs, inexpensive valves and buffers for compression springs.

Robert O. Parmley

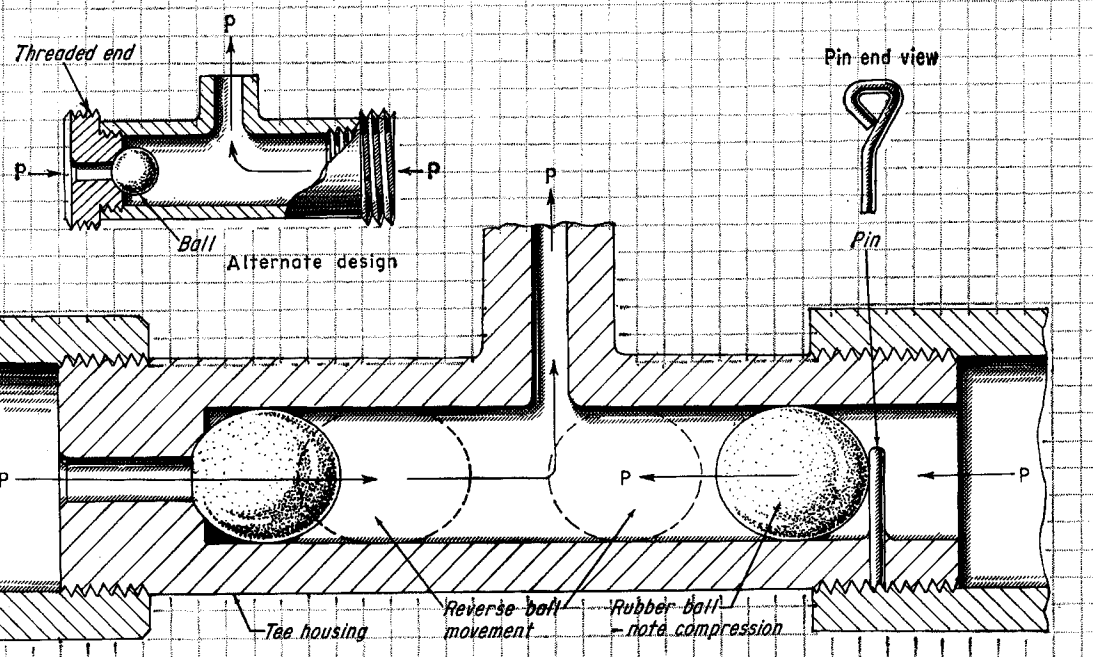




5 CHECKVALVE BALL IS PERMANENTLY INSTALLED



6 SPRING BUFFER FOR COMPRESSION SPRING

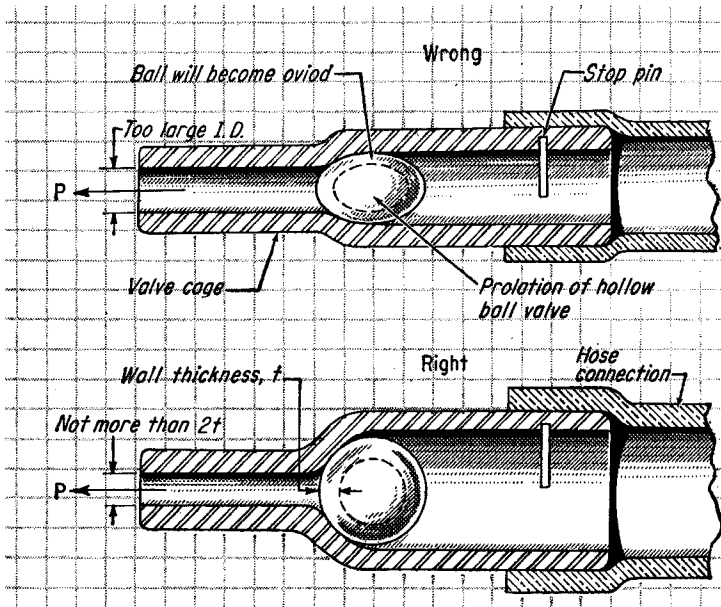


7 TWO-WAY VALVE FOR ALTERNATING PRESSURE

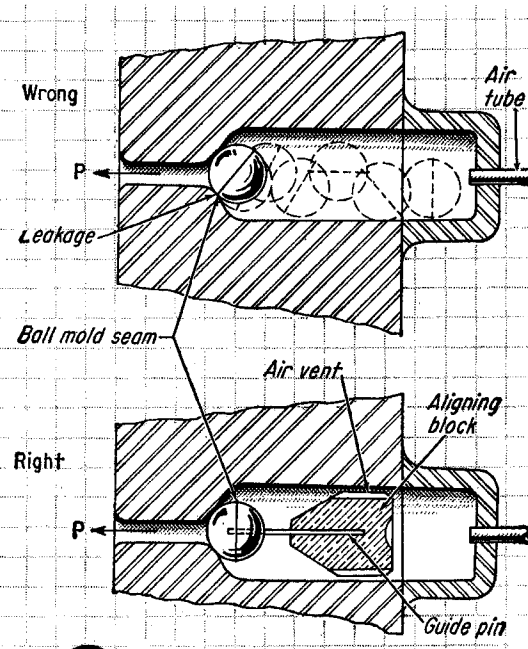
Rubber Balls Find Many Jobs

Plastic and rubber balls, whether solid or hollow, can find a variety of important applications in many designs.

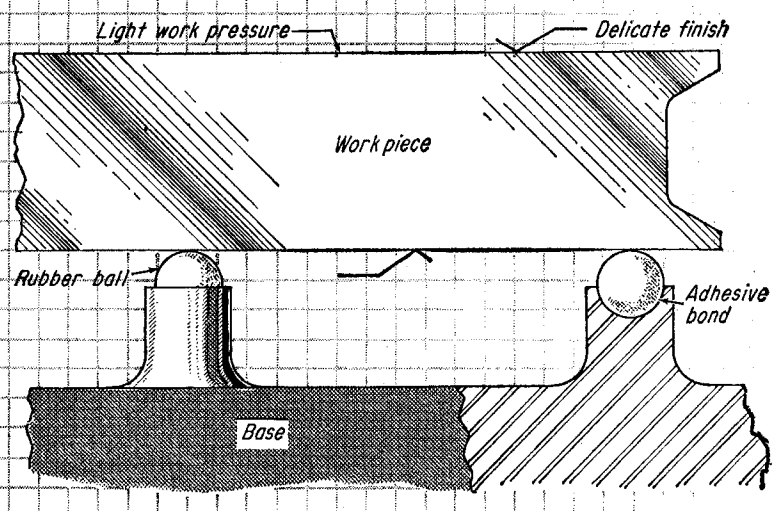
Robert O. Parmley



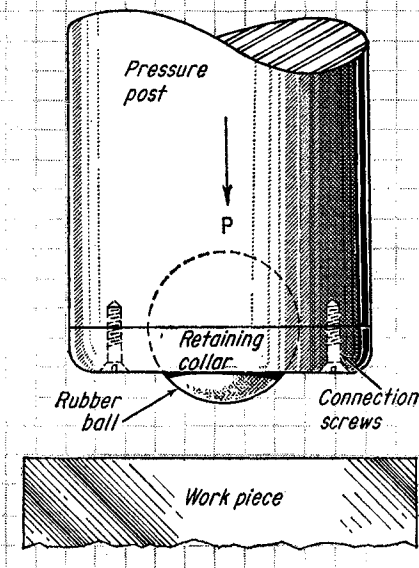
1 **CENTRIFUGALLY MOLDED SPHERES** can be used in efficient, low-cost sealing valves, but avoid ball distortion by making sure that the correct dimensions are applied.



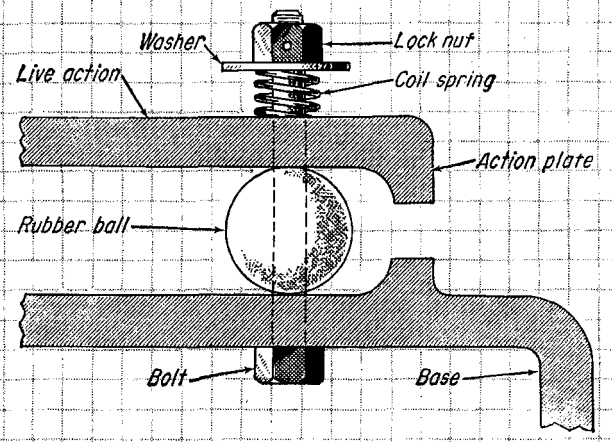
2 **MOLD SEAMS** on solid balls should be held normal to flow line to avoid incomplete seating and consequent leakage.



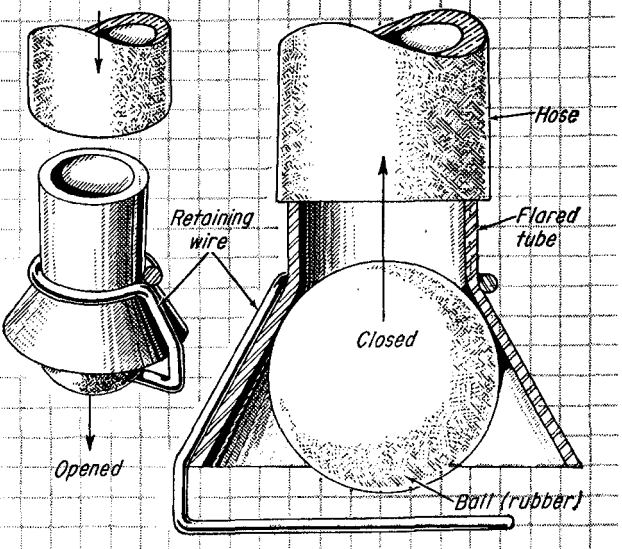
5 **ALIGN DELICATE WORKPIECES** on rubber balls that are bonded into base pillars. Adequate protection of fine finishes is provided, while at the same time friction provides firm grip.



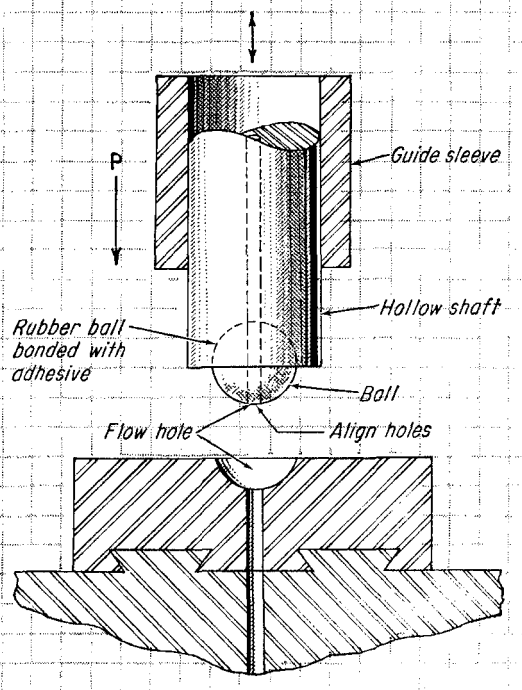
6 **VERTICAL PRESSURE-POST** holds solid ball in easily removed retaining collar. Ball is solid and protects workpiece finishes during assembly operations.



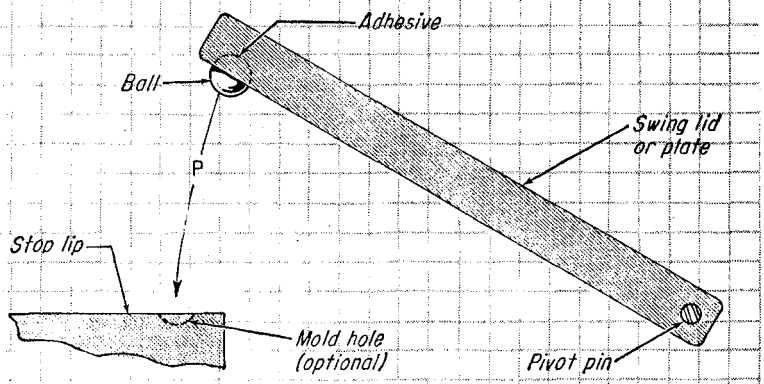
3 **CUSHION PAD AND SHOCK ABSORBER** is easily made for mounting-plates that must either carry dynamic loads or absorb shock forces.



4 **DISCHARGE VALVE** as shown here is an effective way of controlling liquid displacement at the end of hoses where suction would be unwanted or harmful.



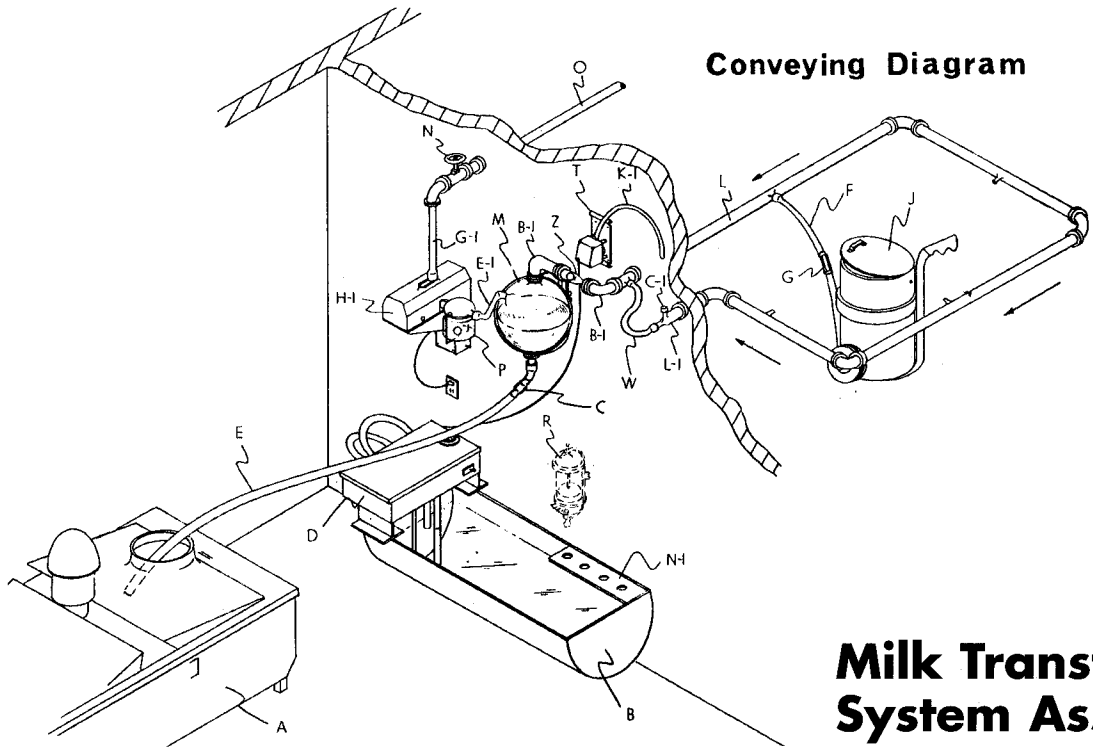
7 **HOLLOW SHAFT-SEAL** embodies adhesive-bonded rubber ball with flow hole. Quick connection of leakproof joint for lubricant or other liquid is gained.



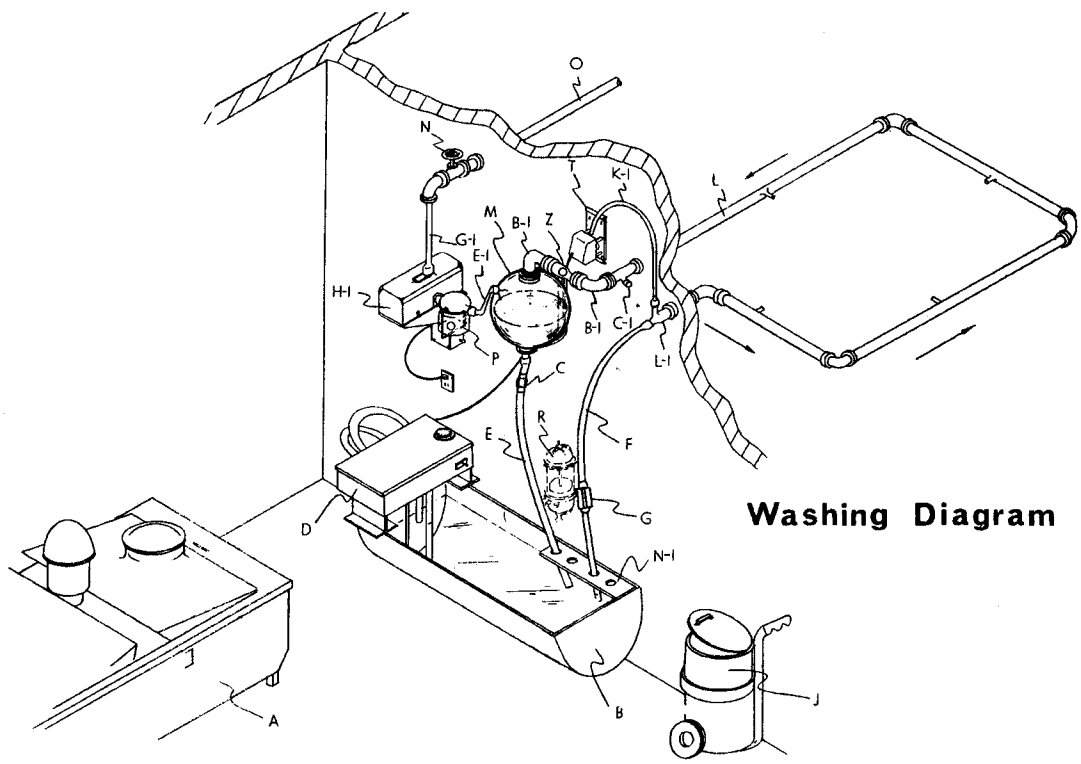
8 **BUMPER STOP** is another example of the simple but effective way a rubber ball can be employed to protect surfaces or parts.

Multiple Use of Balls in Milk Transfer System

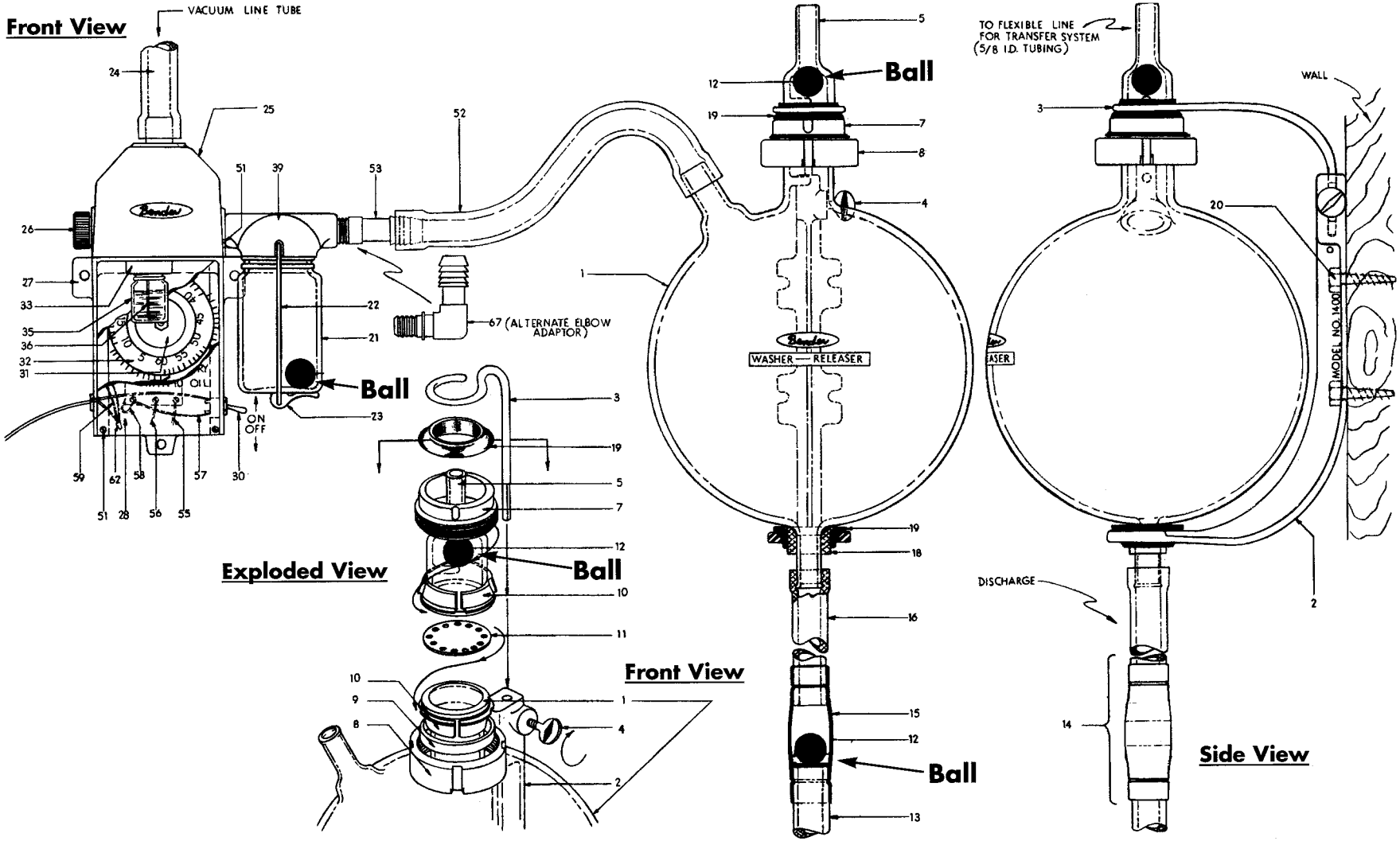
Source: Bender Machine Works, Inc. Illustrated by: Robert O. Parmley



Milk Transfer System Assembly



Washing Diagram



Four plastic balls, located at key positions within the system, act a positive check valves as they respond to the vacuum pulsations.

Use of Balls in Reloading Press

Inventor: E. E. Lawrence Draftsman: R. O. Parmley

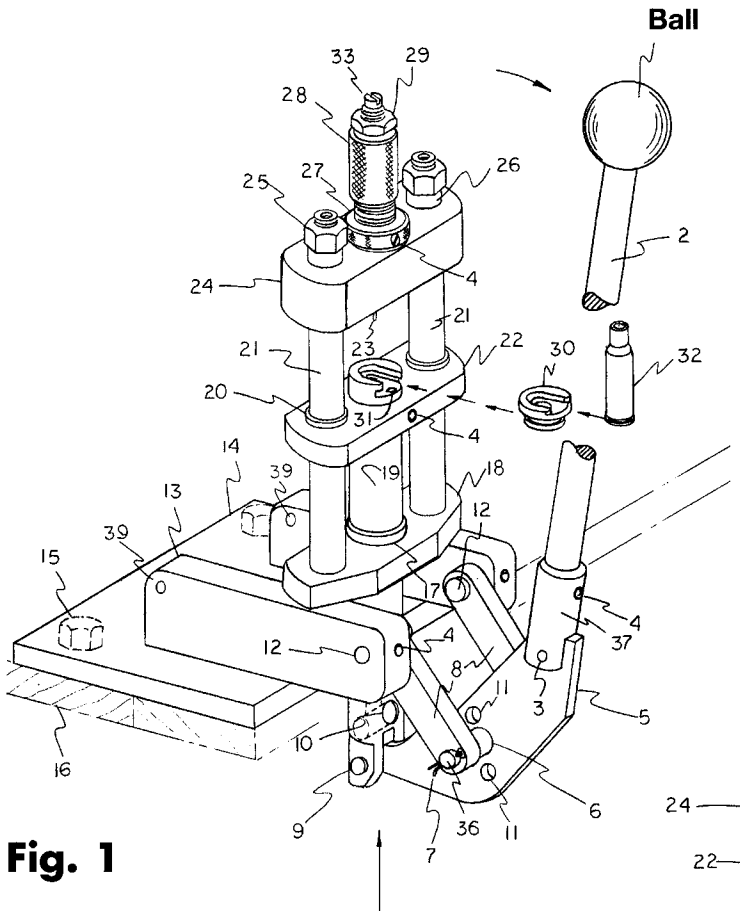


Fig. 1

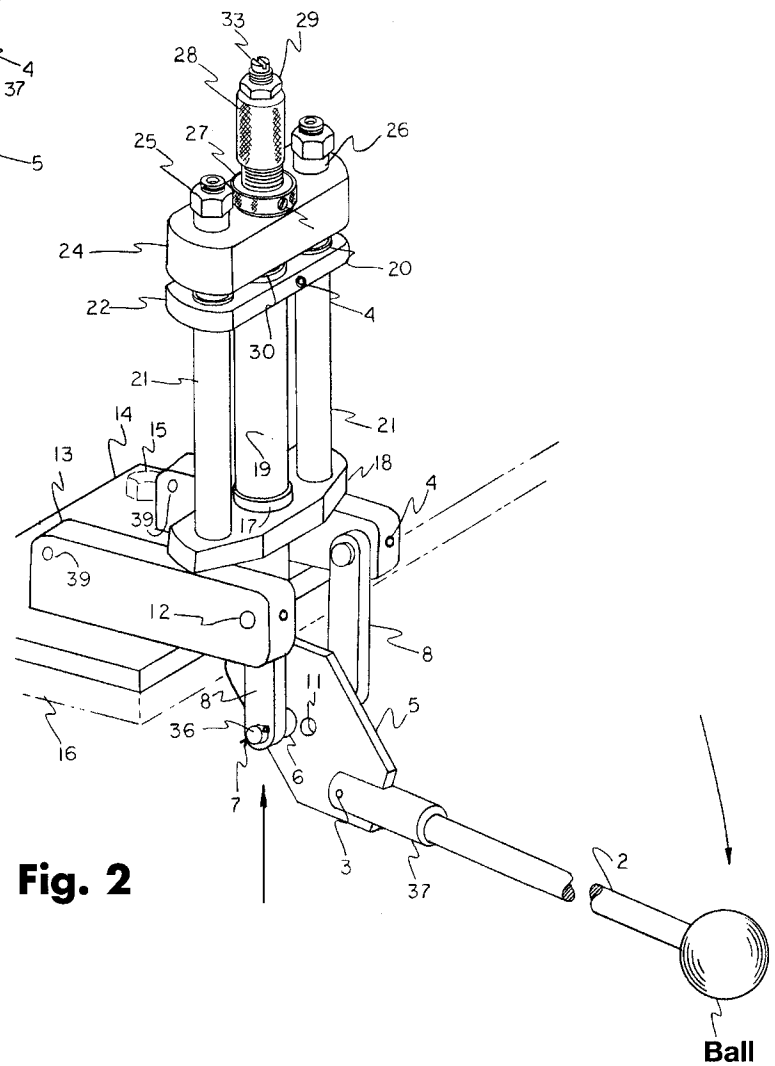


Fig. 2

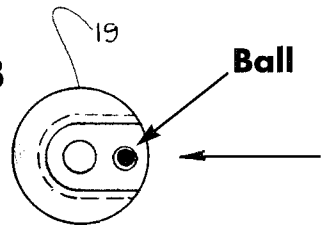


Fig. 3

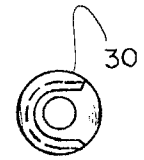
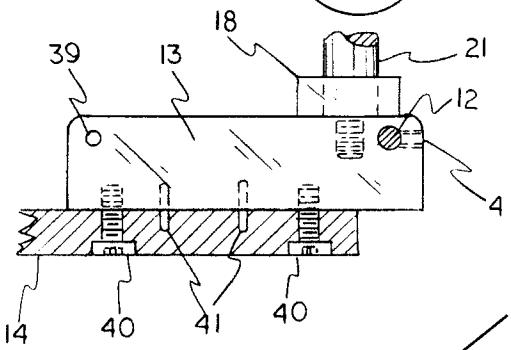


Fig. 4



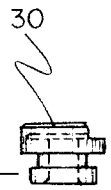


Fig. 5

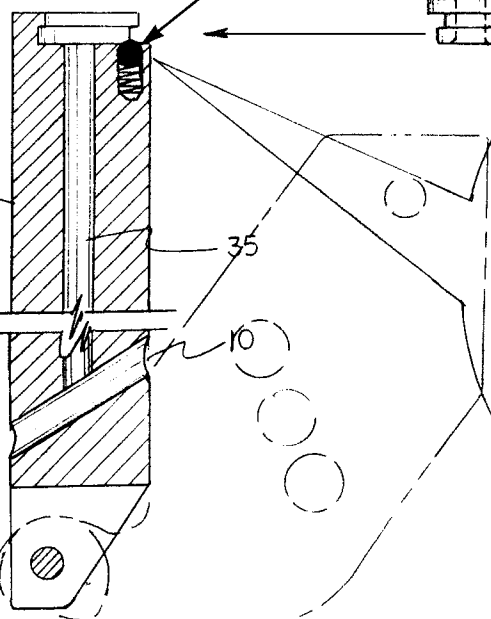


Fig. 6

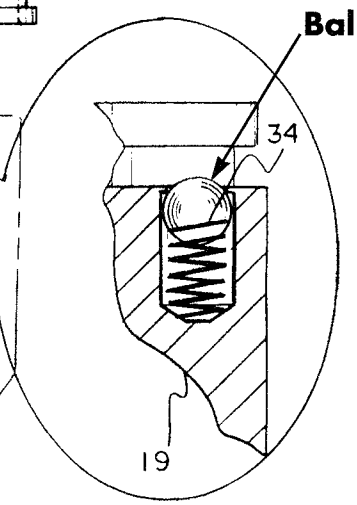
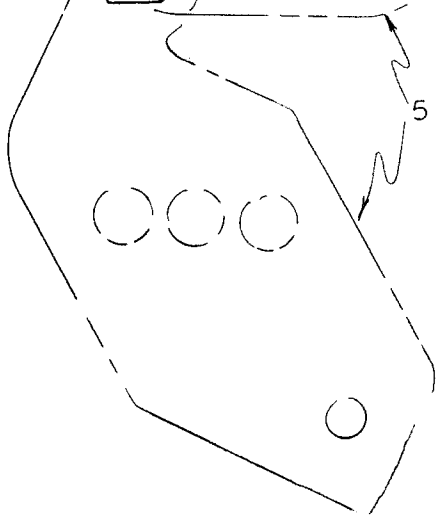
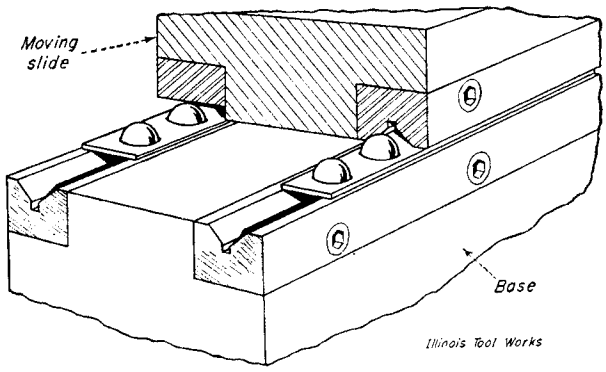


Fig. 8

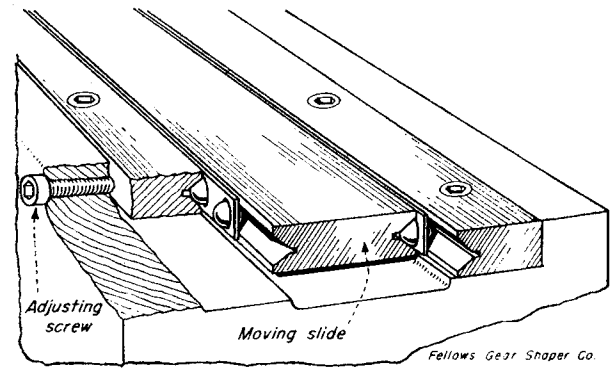
Fig. 7



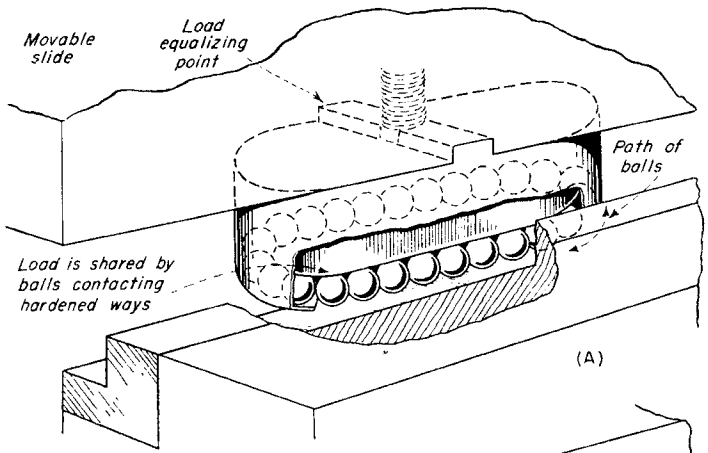
Nine Types of Ball Slides for Linear Motion

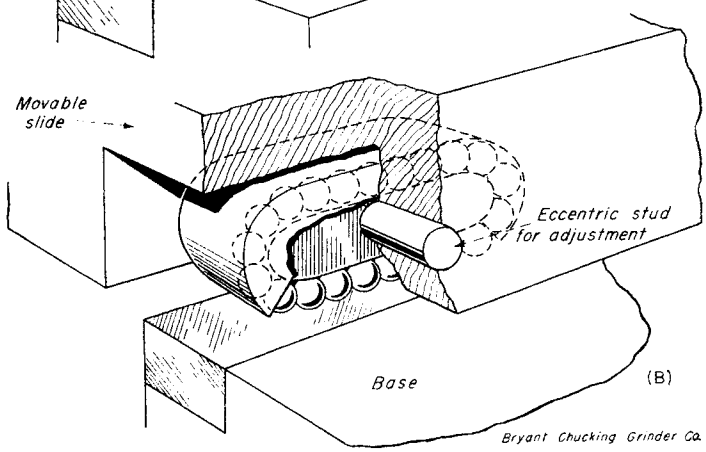


1 V grooves and flat surface make simple horizontal ball slide for reciprocating motion where no side forces are present and a heavy slide is required to keep balls in continuous contact. Ball cage insures proper spacing of balls; contacting surfaces are hardened and lapped.

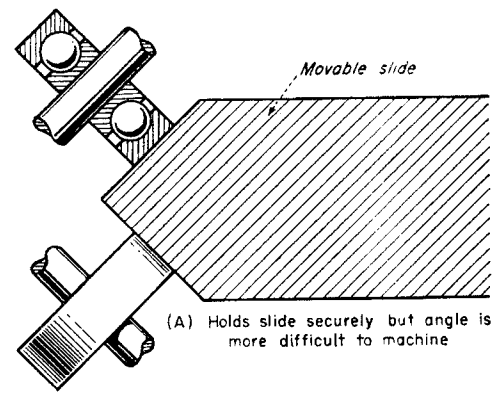


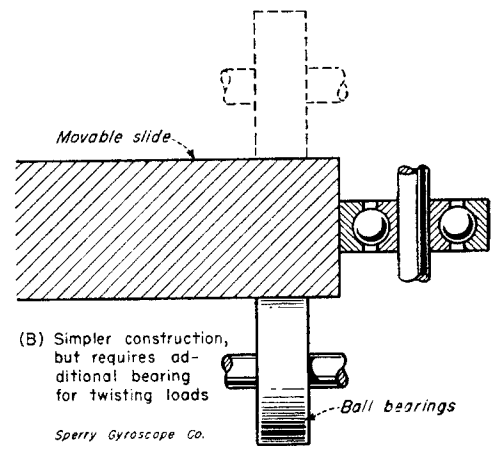
2 Double V grooves are necessary where slide is in vertical position or when transverse loads are present. Screw adjustment or spring force is required to minimize looseness in the slide. Metal-to-metal contact between the balls and grooves insure accurate motion.



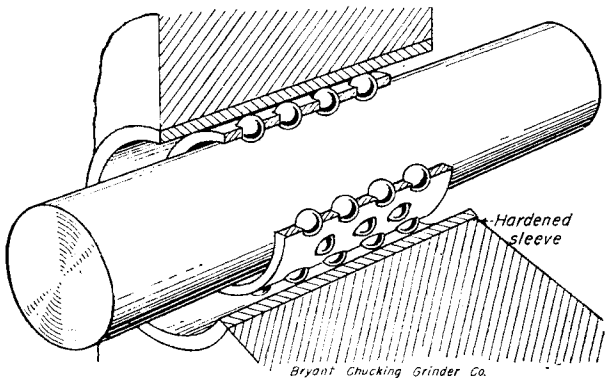


3 Ball cartridge has advantage of unlimited travel since balls are free to recirculate. Cartridges are best suited for vertical loads. (A) Where lateral restraint is also required, this type is used with a side preload. (B) For flat surfaces cartridge is easily adjusted.

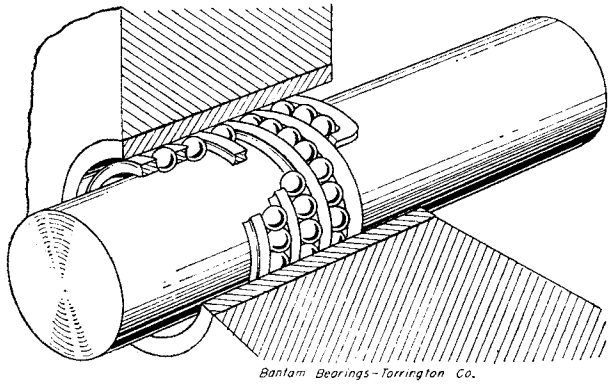



Movable slide
(B) Simpler construction, but requires additional bearing for twisting loads
Ball bearings
Sperry Gyroscope Co.

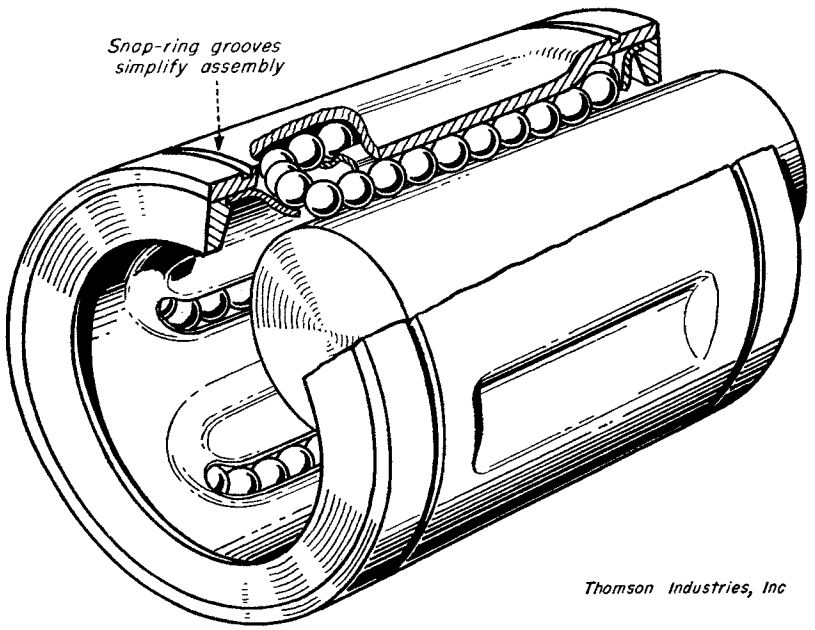
4 Commercial ball bearings can be used to make a reciprocating slide. Adjustments are necessary to prevent looseness of the slide. (A) Slide with beveled ends, (B) Rectangular-shaped slide.



5 Sleeve bearing consisting of a hardened sleeve, balls and retainer, can be used for reciprocating as well as oscillating motion. Travel is limited similar to that of Fig. 6. This type can withstand transverse loads in any direction.



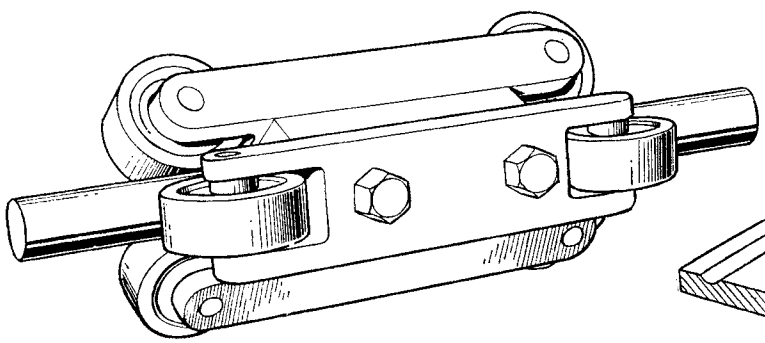
6 Ball reciprocating bearing is designed for rotating, reciprocating or oscillating motion. Formed-wire retainer holds balls in a helical path. Stroke is about equal to twice the difference between outer sleeve and retainer length.

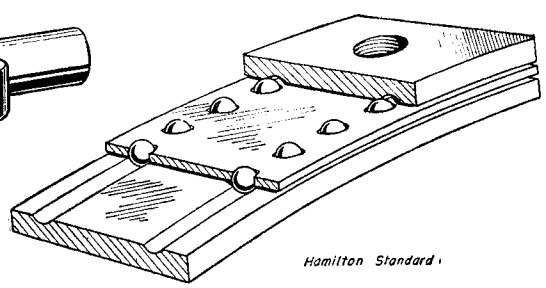


7 Ball bushing with several recirculating systems of balls permit unlimited linear travel. Very compact, this bushing simply requires a bored hole for installation. For maximum load capacity a hardened shaft should be used.

8 Cylindrical shafts can be held by commercial ball bearings which are assembled to make a guide. These bearings must be held tightly against shaft to prevent looseness.

9 Curvilinear motion in a plane is possible with this device when the radius of curvature is large. However, uniform spacing between grooves is important. Circular - sectioned grooves decrease contact stresses.





Unusual Applications of Miniature Bearings

R. H. Carter

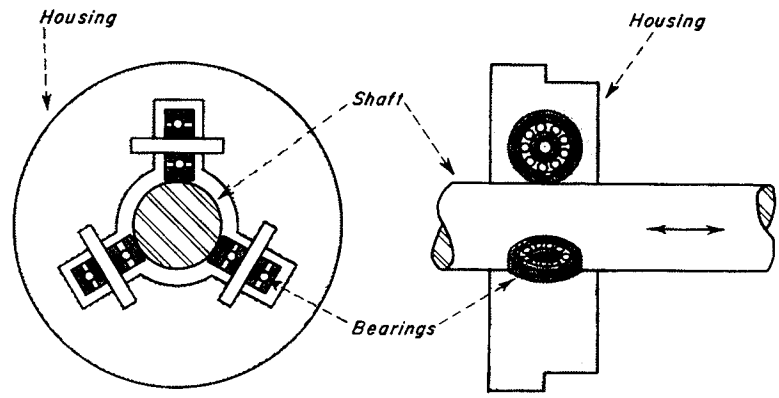


Fig. 1—BALL-BEARING SLIDES. Six miniature bearings accurately support a potentiometer shaft to give low-friction straight line motion. In each end housing, three bearings are located 120 deg radially apart to assure alignment and freedom from binding of the potentiometer shaft.

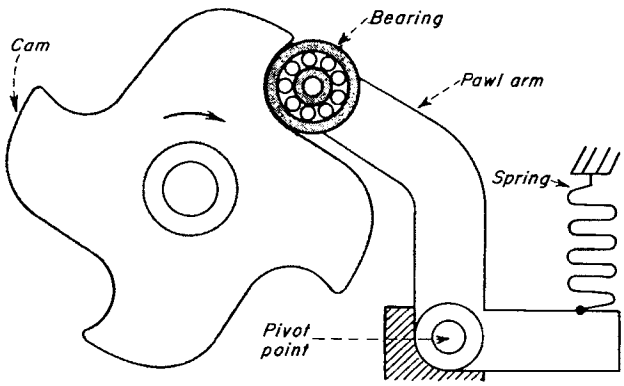


Fig. 2—CAM-FOLLOWER ROLLER. Index pawl on a frequency selector switch uses bearing for a roller. Bearing is spring loaded against cam and extends life of unit by reducing cam wear. This also retains original accuracy in stroke of swing of the pawl arm.

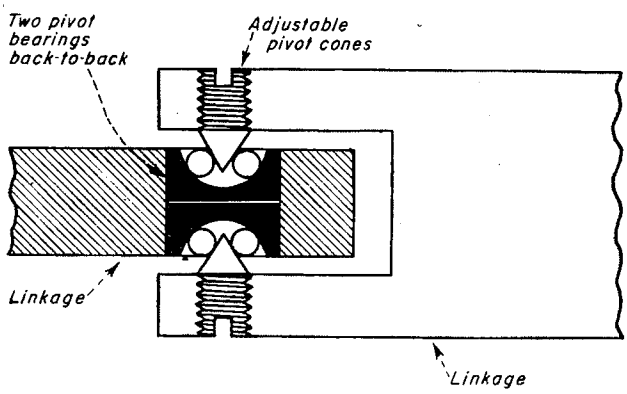


Fig. 3—SEAT FOR PIVOTS. Pivot-type bearings reduce friction in linkages especially when manually operated such as in pantographing mechanisms. Minimum backlash and maximum accuracy are obtained by adjusting the threaded pivot cones. Mechanism is used to support diamond stylus that scribes sight lines on the lenses of gunnery telescopes.

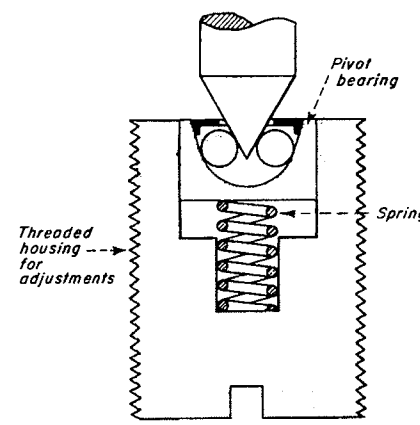


Fig. 4—SHOCK-ABSORBING PIVOT POINT. Bearing with spherical seat resting on spring acts as a pivot point and also absorbs mild shock loads. Used on a recording potentiometer that is temperature controlled. Spring applies uniform load over short distances and gives uniform sensitivity to the heat-sensing element. Close fit of bearing in housing is required.

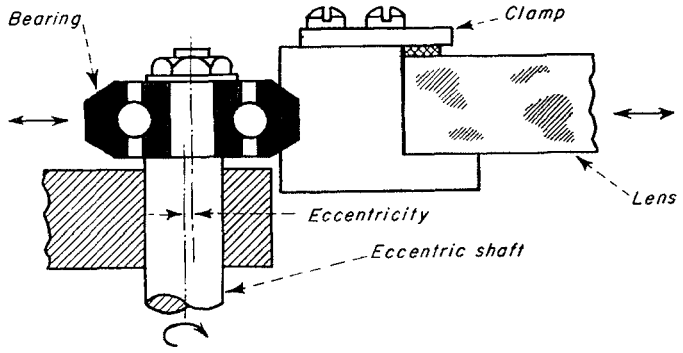


Fig. 5—PRECISE RADIAL ADJUSTMENTS obtained by rotating the eccentric shaft thus shifting location of bearing. Bearing has special-contoured outer race with standard inner race. Application is to adjust a lens with grids for an aerial survey camera.

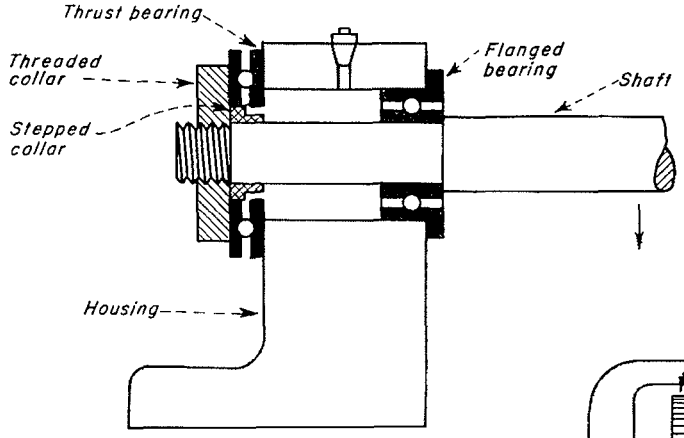


Fig. 6—SUPPORT FOR CANTILEVERED SHAFT obtained with combination of thrust and flanged bearings. Stepped collar provides seat for thrust bearing on the shaft but does not interfere with stationary race of thrust bearing when shaft is rotating.

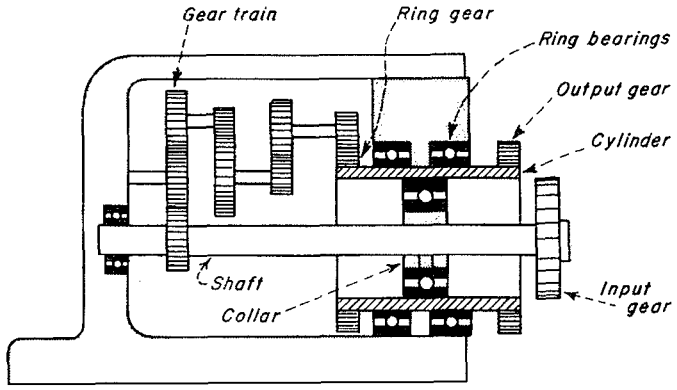


Fig. 7—GEAR-REDUCTION UNIT. Space requirements reduced by having both input and output shafts at same end of unit. Output shaft is a cylinder with ring gears at each end. Cylinder rides in miniature ring bearings that have relative large inside diameters in comparison to the outside diameter.

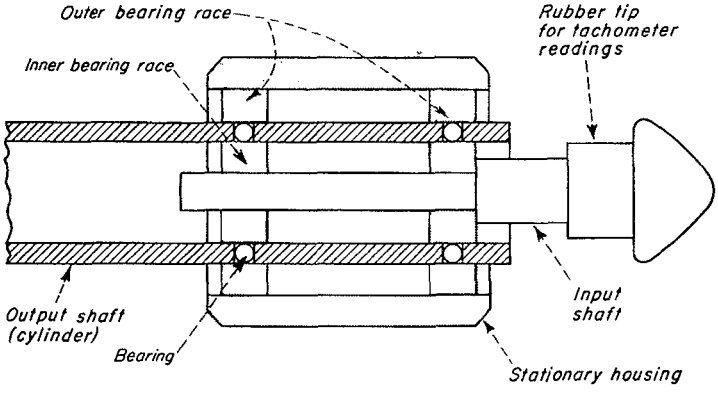


Fig. 8—BEARINGS USED AS GEARS. Manually operated tachometer must take readings up to 6000 rpm. A 10-to-1 speed reduction was obtained by having two bearings function both as bearings and as a planetary gear system. Input shaft rotates the inner race of the inner bearings, causing the output shaft to rotate at the peripheral speed of the balls. Bearings are preloaded to prevent slippage between races and balls. Outer housing is held stationary. Pitch diameters and ball sizes must be carefully calculated to get correct speed reduction.

Rolling Contact Bearing Mounting Units

FIG. 1—Pillow blocks are for supporting shafts running parallel to the surface on which they are mounted. Provision for lubrication and sealing are incorporated in the pillow block unit. Assembly and disassembly are easily accomplished. For extremely precise installations, mounting units are inadvisable.

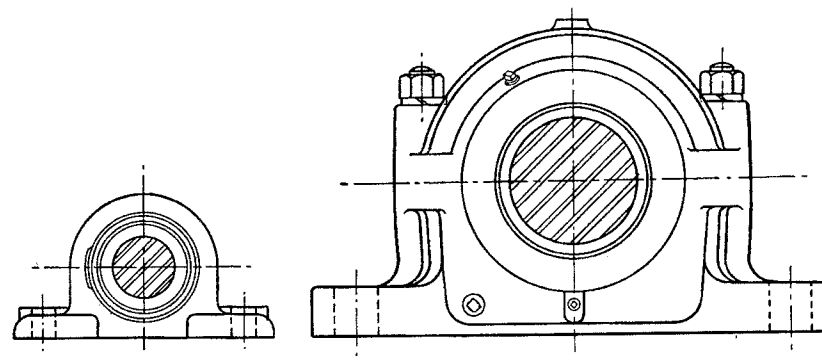


FIG. 2—Pillow blocks can be designed to prevent the transmission of noise to the support. One design (A) consists of a bearing mounted in rubber. The rubber in turn is firmly supported by a steel casing. Another design (B) is made of synthetic rubber. Where extra rigidity is required the synthetic rubber mount can be reinforced by a steel strap bolted around it.

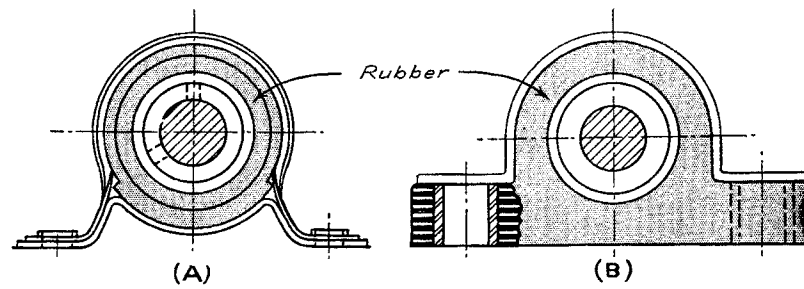
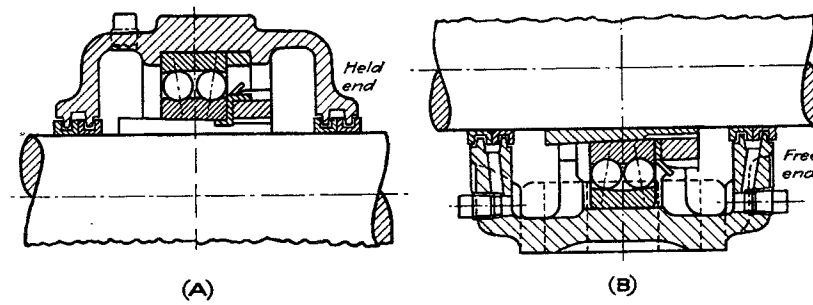


FIG. 3—Changes in the temperature are accompanied by changes in the length of a shaft. To compensate for this change in length, the pillow block (B) supporting one end of the shaft is designed to allow the bearing to shift its position. The pillow block (A) at the other end should not allow for longitudinal motion.



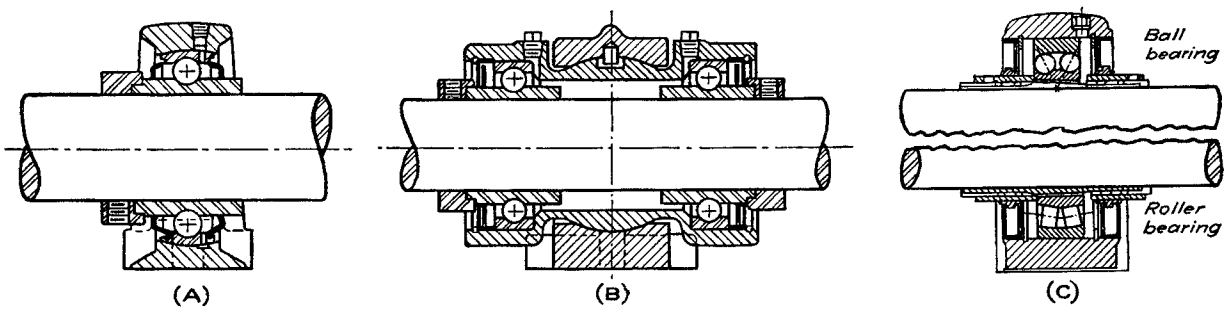


Fig. 4—Compensation for misalignment can be incorporated into pillow blocks in various ways. One design (A) uses a spherical outer surface of the outer race. Design (B) uses a two-part housing. The spherical joint compensates for misalignment. Another design (C) uses a spherical inner surface of the outer race.

Fig. 5—The cylindrical cartridge is readily adaptable to various types of machinery. It is fitted as a unit into a straight bored housing with a push fit. A shoulder in the housing is desirable but not essential. The advantages of a predesigned and preassembled unit found in pillow blocks also apply here.

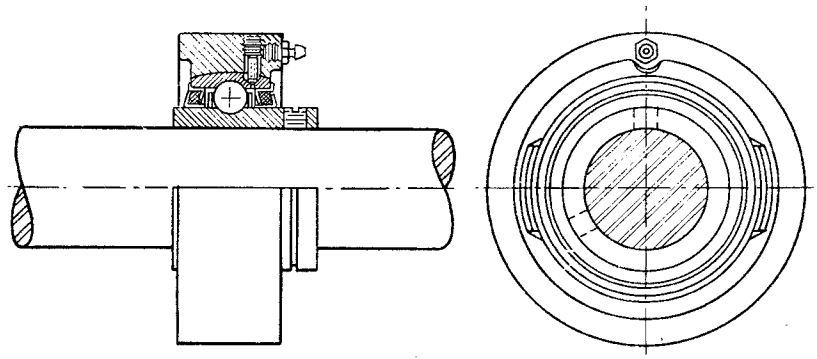


FIG. 6—The flange mounting unit is normally used when the machine frame is perpendicular to the shaft. The flange mounting unit can be assembled without performing the special boring operations required in the case of the cartridge. The unit is simply bolted into the housing when it is being installed.

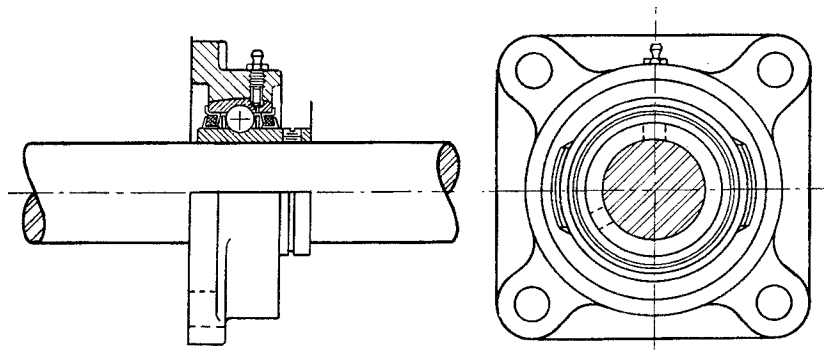
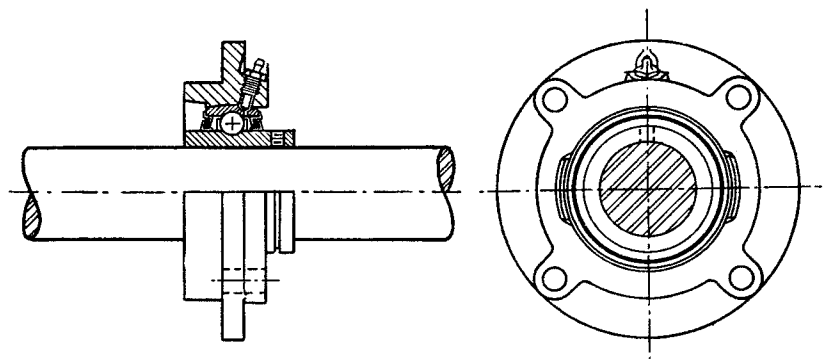


FIG. 7—The flange cartridge unit projects into the housing and is bolted in place through the flange. The projection into the housing absorbs a large part of the bearing loads. A further use of the cylindrical surface is the location of the mounting unit relative to the housing.



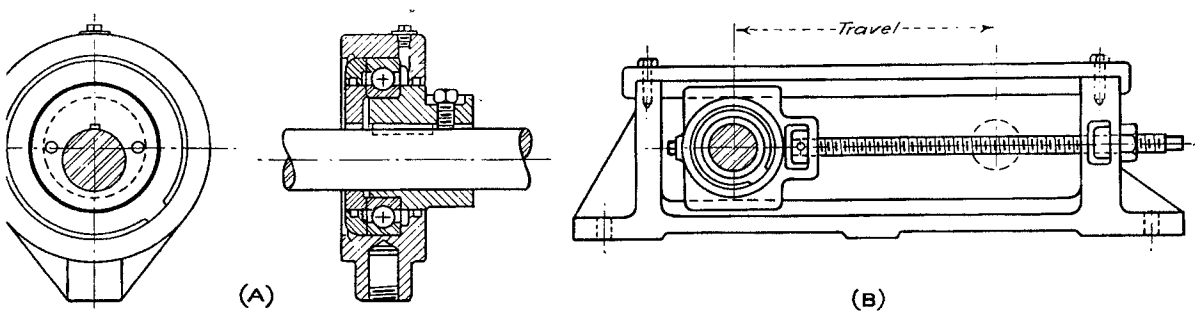


FIG. 8—Among specialized types of mounting units are (A) Eccentrics used particularly for cottonseed oil machinery and mechanical shakers and (B) Take-up units which make possible an adjustment in the position of the shaft for conveyor units. Many other types of special rolling contact bearing mounting units are made.

Eleven Ways to Oil Lubricate Ball Bearings

D. L. Williams

The method by which oil should be applied to a ball bearing depends largely on the surface speed of the balls. Where ball speeds are low, the quantity of oil present is of little importance, provided it is sufficient. Over-lubrication at low speeds is not likely to cause any serious temperature rise. However, as speeds increase, fluid friction due to churning must be avoided. This is done by reducing the amount of oil supplied and by having good drainage from the housing. At very high speeds, with light loads, the oil supply can be limited to a very fine mist.

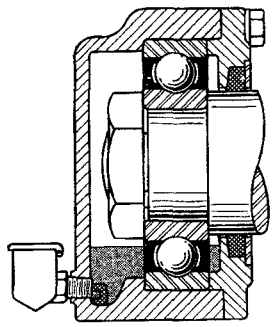


FIG 1

Fig. 1—*Oil Level System.* For moderate speeds, the bearing housing should be filled with oil to the lowest point of the bearing inner race. An oil cup is located to maintain this supply level. Wick acts as a filter when fresh oil is added. This system requires periodic attention.

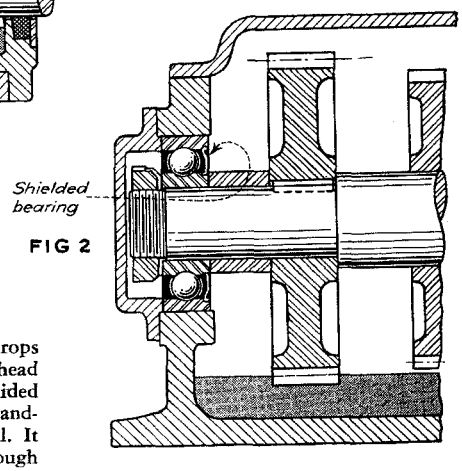


FIG 2

Fig. 2—*Splash Feed* is used where rotating parts require oil for their own lubrication. Splash lubrication is not recommended for high speeds because of possible churning. Bearings should be protected from chips or other foreign material by using a shaft mounted slinger or shielded bearings.

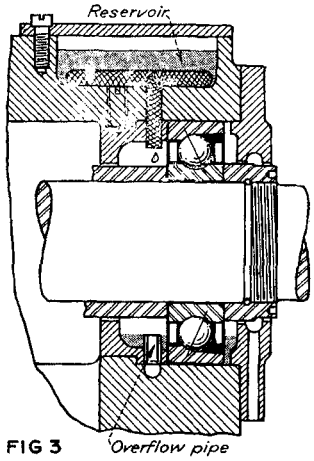


FIG 3

Fig. 3—*Drop Feed.* Oil may be fed in drops using either sight-feed oilers or an overhead reservoir and wick. Drains must be provided to remove excess oil. A short overflow standpipe, serves to maintain a proper oil level. It also retains a small amount of oil even though the reservoir should be empty.

Fig. 4—*Spray Feed.* With higher speeds, definite control of oil fed to bearings is important. This problem is more difficult for vertical bearings because of oil leakage. One method uses a tapered slinger to spray oil into the bearings. Oil flow is altered by the hole diam., the taper and oil viscosity.

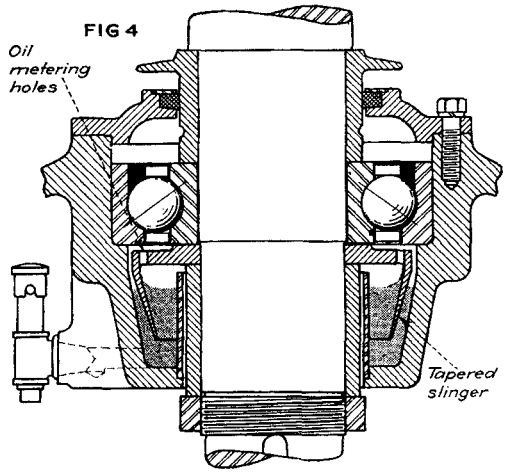


FIG 4

Fig. 5—*Circulating Feed.* Most circulating systems are somewhat complicated and expensive but this is justified by their permanence and reliability. Oil reservoir is attached to the shaft and when rotated, the oil is forced upward where it strikes a scoop, flows through and onto the bearing.

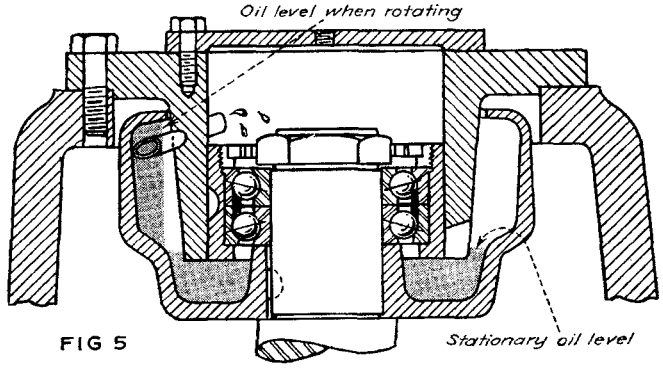


FIG 5

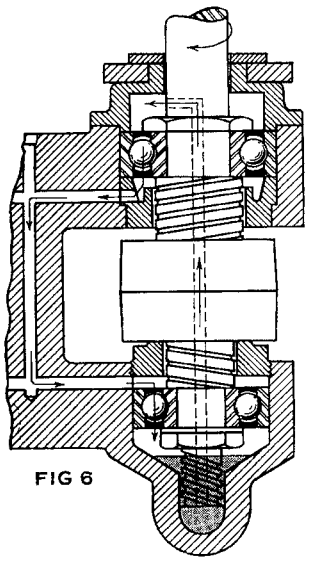


Fig. 6—*Most circulating systems* are used for vertical shaft applications and usually where ball speeds are comparatively high. One system consists of an external screw which pumps the oil upward through the hollow spindle to a point above the top bearings.

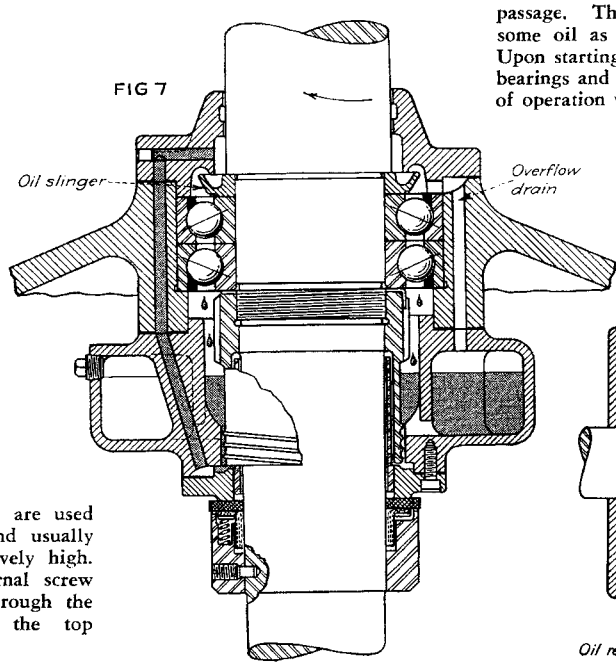


Fig. 7—*Another screw pump application* forces the oil upward through an external passage. The cup-shaped slinger traps some oil as the spindle comes to rest. Upon starting, this oil is thrown into the bearings and avoids a short initial period of operation with dry bearings.

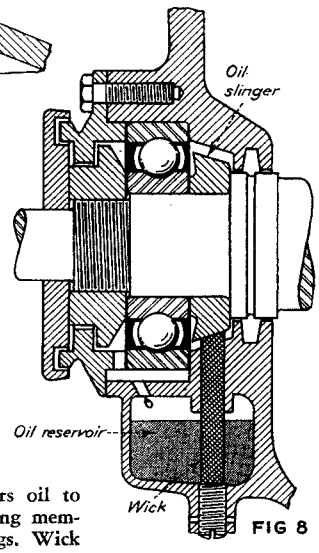


Fig. 8—*Wick Feed* filters and transfers oil to a smoothly finished and tapered rotating member which sprays a mist into bearings. Wick should be in light contact with the slinger or else the wick may become glazed or charred. A light spring is often used for proper wick pressure.

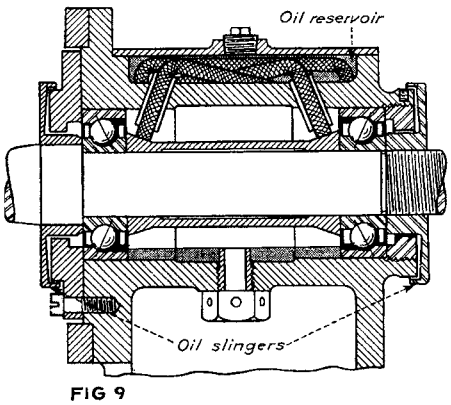


Fig. 9—*Wick feeds* are used in applications of extremely high speeds with light loads and where a very small quantity of oil is required in the form of a fine mist. Slingers clamped on the outside tend to draw the mist through the bearings.

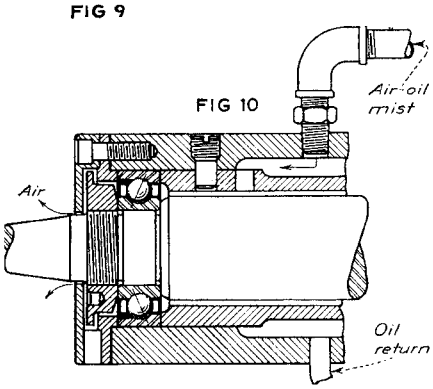


Fig. 10—*Air-Oil Mist.* Where the speeds are quite high and the bearing loads relatively light, the air-oil mist system has proven successful in many applications. Very little oil is consumed and the air flow serves to cool bearings.

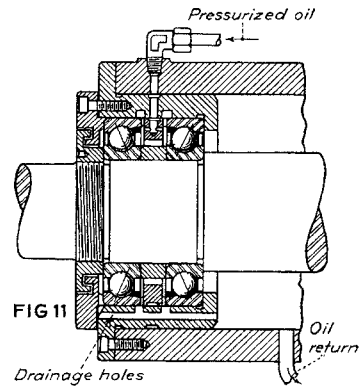


Fig. 11—*Pressure Jet.* For high speeds and heavy loads, the oil must often function as a coolant. This method utilizes a solid jet of cool oil which is directed into the bearings. Here adequate drainage is especially important. The oil jets may be formed integrally with the outer preload spacer.

Ball-Bearing Screw Life

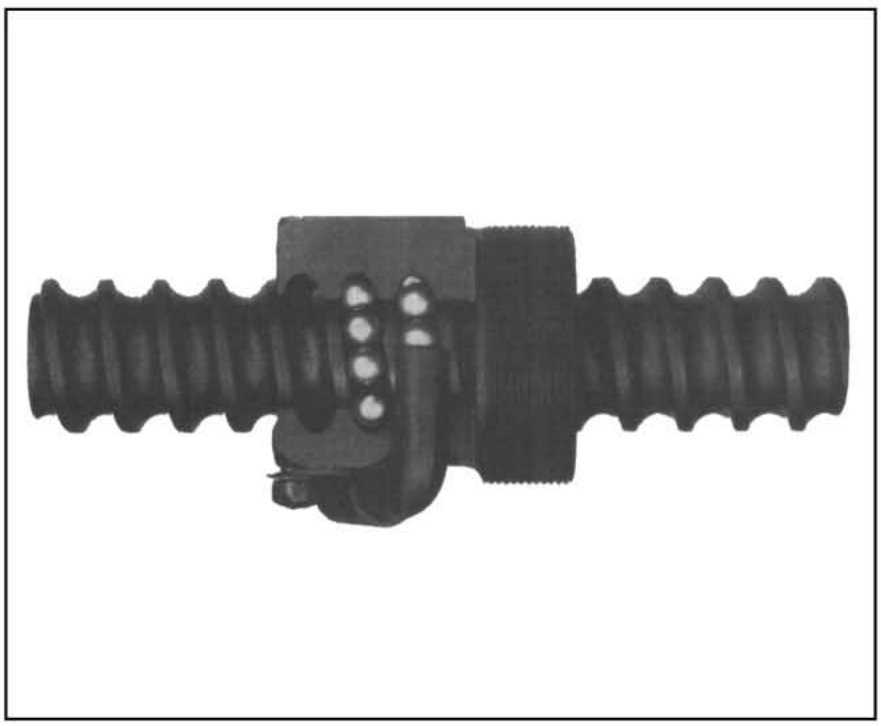
Based on several years of life-load tests, the charts reduce calculation time and increase reliability-analysis.

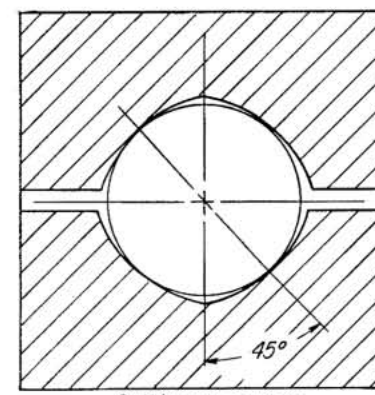
D. A. Galonska

BALL-BEARING screws are designed individually to meet the specific load, life and space requirements of each application. To simplify such calculations and to provide a higher degree of reliability, a series of tests was conducted which furnished data for the construction of two design charts. With one—a life-load chart—minimum life of ball-bearing screws can now be predicted to within 10 percent. A second chart aids in determining the key dimensions. A numerical problem is included here to illustrate the use of the charts. Also given are rules for selecting the optimum number of balls and circuits in a unit, best materials, hardnesses.

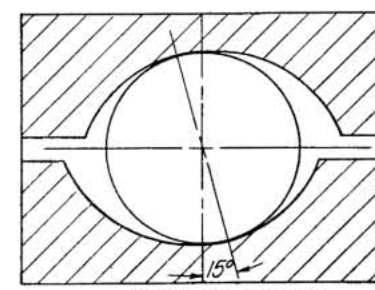
Basic types

Ball-bearing screws are highly efficient units for precision positioning with minimum power, size and cost. With mechanical efficiencies ranging from 90 to 95 percent, ball-bearing screws require only one-third as much torque for the same linear output as conventional Acme thread screws.





Gothic arc grooves



Circle-arc grooves

BASIC BALL-BEARING screw assembly consists of a screw and nut having helical races separated by balls. Two popular types of grooves (right).

The basic unit of a ball-bearing screw assembly consists of a screw and nut having helical races separated by balls. A tubular guide on the nut interrupts the path of the balls, deflects them from the races, and guides them diagonally across the outside of the nut and back to the races. In operation, the rolling balls recirculate continuously through this closed circuit as nut and screw rotate in relation to each other.

The lead of a ball-bearing screw is the distance the nut (or screw) advances for one revolution of the screw (or nut). It is usually expressed as a decimal dimension, but may be given in threads per inch. The ball circle diameter, or pitch diameter, is the diameter of a circle whose radius is the distance from the screw axis to the center of the active bearing balls.

Grooves forming the helical races of ball-bearing screws and nuts may be either of circle arc or Gothic arc cross-section. The Gothic arc groove design minimizes lash by reducing the axial freedom of the assemblies. Also, with this construction, foreign matter entering the grooves is pushed by the balls into the space at the apex. The design of the Gothic arc groove shape is usually based on a 45-degree contact angle, while with circular grooves, the contact angle varies with changes in load, lash, and ball size. The circular groove design, however, may offer a slightly lower frictional loss during operation.

Load-carrying capacity

Load capacity depends on material, hardness, ball and screw diameters and on the number of bearing balls. However, screw and ball diameters are generally limited by the lead specified or space available; hence, to increase the load capacity, it is usually necessary to increase the number of balls. If too many balls or too many turns are designed in a single long circuit, there is a tendency to jam or lock because of the friction caused by the rubbing of adjacent balls rolling in the same direction.

One way to reduce the tendency to jam is to include alternate balls of a smaller diameter. The larger ones serve as bearing balls, the smaller ones as spacers. In this way, adjacent balls rotate in opposite directions, similar to idler gears in a gear train. Obviously this design carries less load for a given space and weight than types in which all the balls are load carriers.

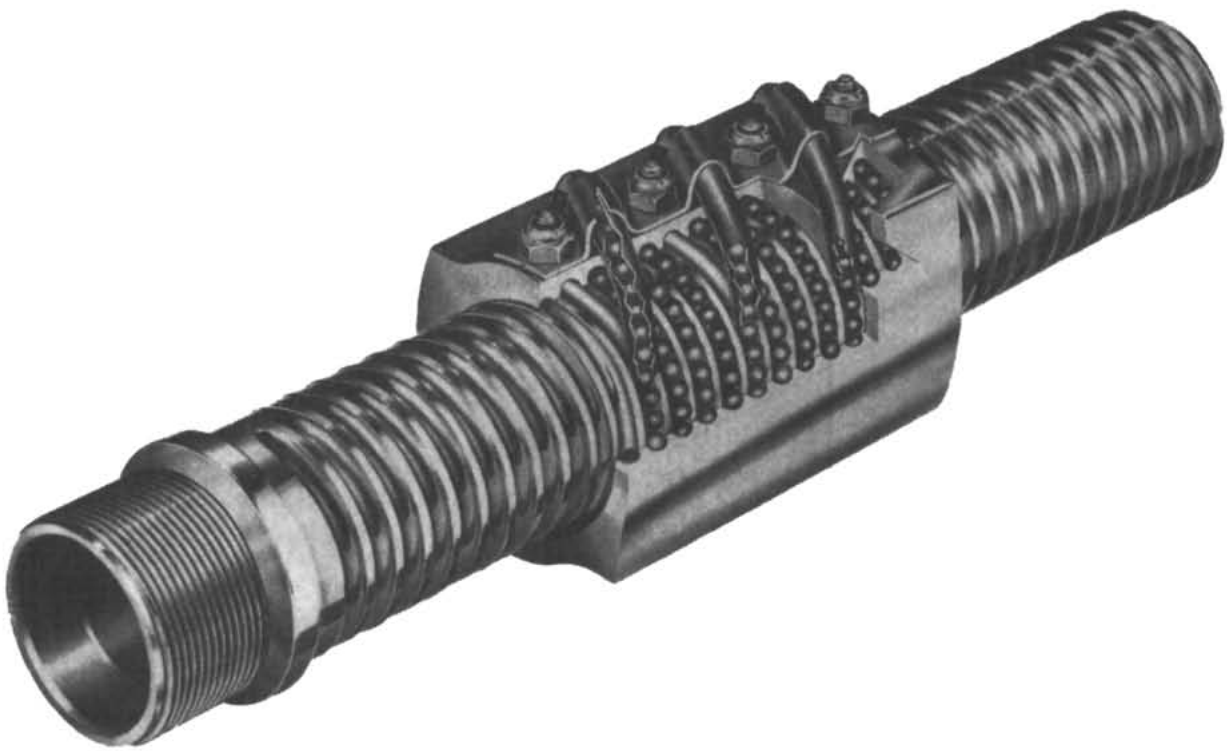
Another method for increasing the number of balls, and thus raising the load-carrying capacity of a ball-bearing nut of given length, is to provide more than one circuit. In a multiple-circuit design, the separate circuits divide the load equally. Also, every ball is a load carrier, and the need for extra nonworking spacer balls is eliminated. Another important advantage is that if one circuit fails, the others can generally carry the load until repairs can be made.

Tests have determined two limiting factors when all balls are to be load carriers:

1. Number of balls in any single circuit should be less than 125.
2. Maximum circuit length should not exceed 3½ turns.

Little is gained by providing more circuits having fewer turns. In one series of tests it was found that the life of nuts having two circuits of 3½ turns each was comparable to that of a nut having five circuits of 1½ turns each.

Load-carrying capacity of ball-bearing screws closely parallels that of conventional ball bearings. Stress levels and impacts on the races determine the life of an assembly. Stress level (load rating) versus number of impacts (or screw revolutions) have been



MULTIPLE BALL CIRCUITS increase load-carrying capacity. Each circuit carries equal share of load.

have been determined by laboratory test under simulated service conditions, Fig 1 and 2, pp 52-53. The ratings are specified in terms of one million revolutions. Use of the charts is illustrated in the following problem.

Design problem

Design a ball-bearing screw of minimum size and weight to meet the specifications listed below (see also illustration below). The unit is to operate an aircraft hydraulic locking cylinder. Also given are typical limits on dimensions and load.

Given

- Nut rotated by input drive, but prevented from shifting linearly; screw does the driving.
- Life requirement is 5000 cycles (in both directions).
- Stroke is 5 in. under load in one direction; the screw remains under compression during the return stroke. (Units with strokes as much as 50 ft have been designed and tested.
- Load is 9300 lb in both directions. (Units have been built to provide a thrust of 1,000,000 lb.)
- Ball-circle diameter of pitch dia, D is 1.25 in. (manufacturing limits: min = $\frac{5}{16}$ in.; max $\approx$ 8 in.)
- Lead = 0.3125 in./rev. (Leads from 0.125 to 1.5 times the pitch diameter are best, although there is no definite top limit.)
- Ball diameter, $d = \frac{7}{32}$ in. The lead specified, as well as the ball-circle diameter, limit the maximum size of the balls because the lands between the grooves must be sufficiently wide to provide adequate support. Also, a portion of the land on the nut is removed by the counterboring required for the ball return system. In this instance, the maximum ball diameter of $\frac{7}{32}$ in. was dictated by experience.

Compute

Total travel = 5 in. stroke in each direction
= 10 in./cycle
= 10 × 5000
= 50,000 in.

rev/in. = 1/0.3125 = 3.2

Total revolutions = 3.2 × 50,000
= 160,000 rev

$$\text{Diameter ratio} = D/d = \frac{1.25}{7/32}$$

$$= 5.71.$$

(Ideal D/d ratio is between 4 and 8.)

From charts

Number of impacts per revolution for a D/d ratio of 5.71 is 7.8, Fig 2. Impacts are the number of balls that pass one point on the nut in one revolution of the screw. It is best to keep the number of impacts within 5 to 13.6 per revolution. Note from the chart that if the nut were driving, with the screw stationary, the higher diagonal line would be read, resulting in a higher number of impacts.

Multiplying the number of revolutions to be traveled (160,000) by the number of impacts per revolution (7.8), we find the total number of impacts to be 1,248,000. Referring to Fig 1, for this number of impacts and $\frac{7}{32}$ in. dia balls, the load that can be carried per ball is 150 lb. Thus

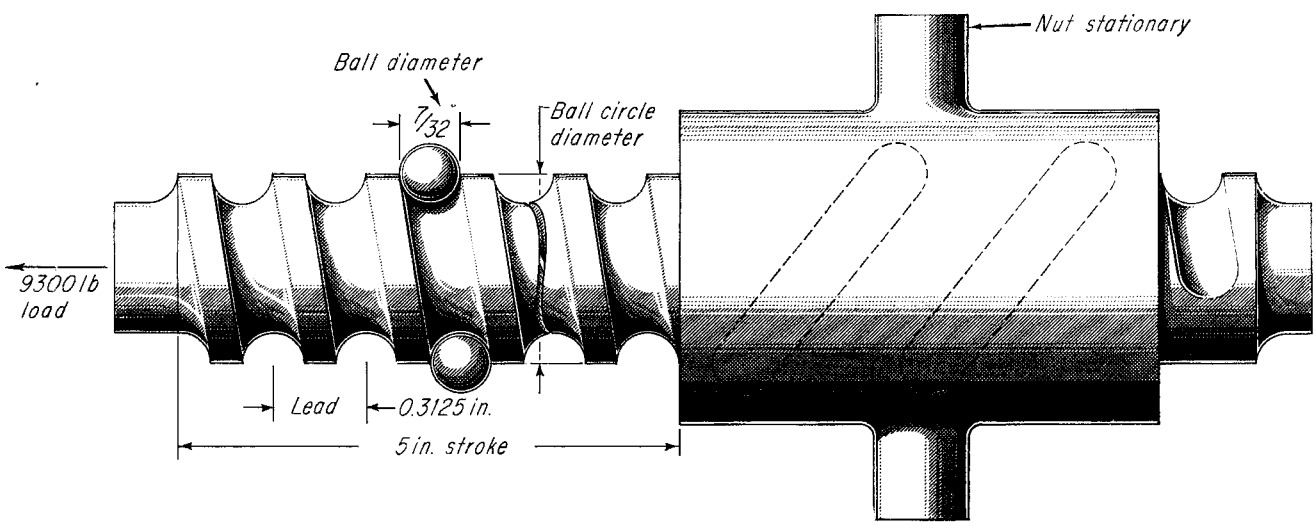
$$\text{No. of balls required} = \frac{9300}{150}$$

$$= 62 \text{ balls.}$$

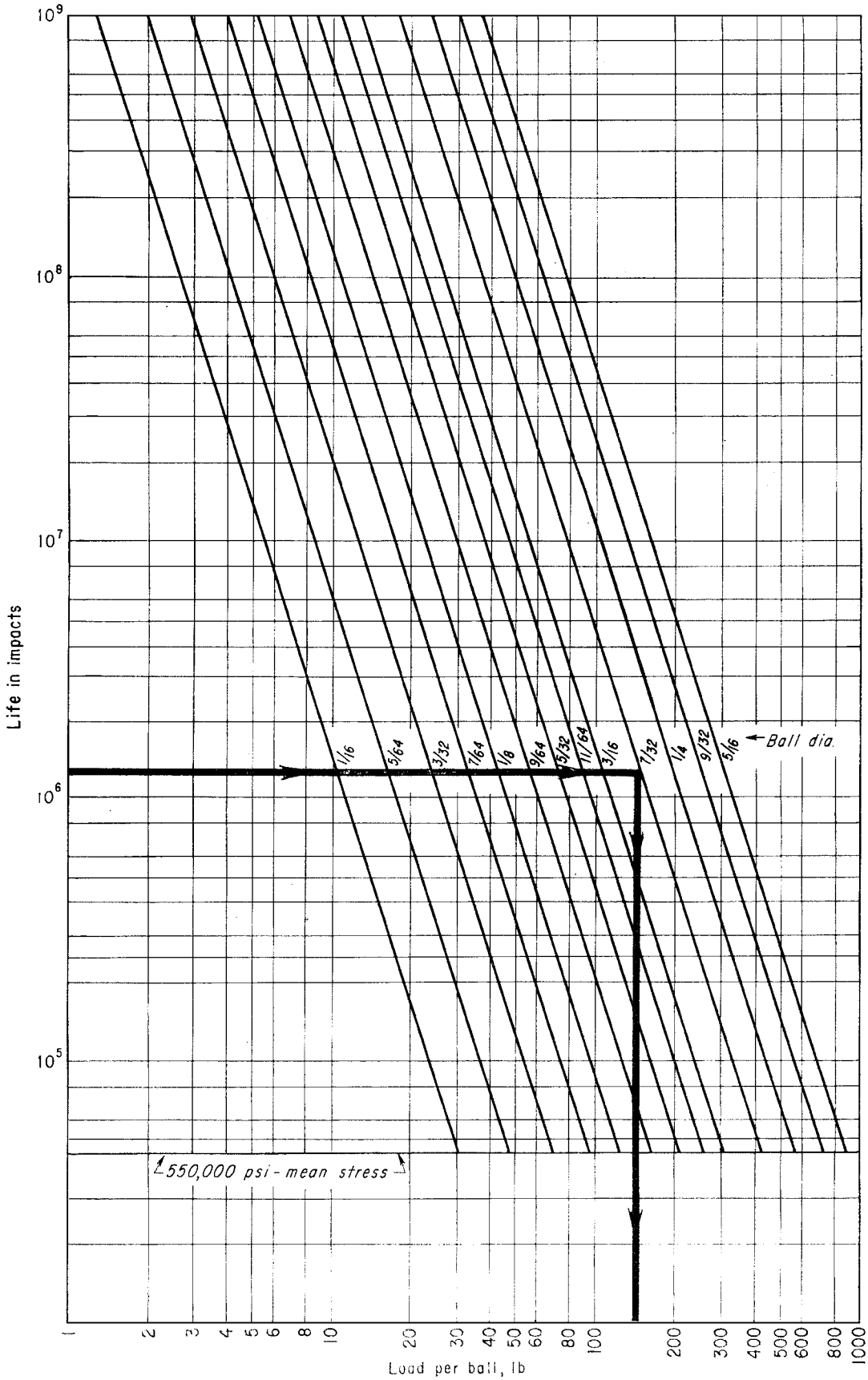
This is less than the maximum of 125 balls per circuit necessary to avoid locking; hence only one circuit is required. If more than 125 balls were required, divide the total by 125 and use the next largest whole number as the number of circuits.

Number of balls per turn is

$$\pi\left(\frac{D}{d}\right) = 5.71\pi = 17.9 \approx 18$$



DIMENSIONS for design problem. Nut rotates, but is stationary in a linear direction.



1 · · LIFE-LOAD RELATIONSHIPS for various diameter balls.

Number of turns is

$$\frac{\text{No. of balls}}{\text{No. of balls per turn}} = \frac{62}{18} = 3.44 = 3\tfrac{1}{2}$$

The number of turns determines the minimum length of nut. In general, the minimum nut length can be approximated from the following table:

TOTAL NUMBER OF TURNS	MINIMUM LENGTH OF BALL NUT
$2\frac{1}{2}$	$4\frac{1}{2} \times$ Lead
$3\frac{1}{2}$	$5\frac{1}{2} \times$ Lead
5	$7\frac{1}{2} \times$ Lead
7	$9\frac{1}{2} \times$ Lead
$10\frac{1}{2}$	$13\frac{1}{2} \times$ Lead

Effect of a varying load

In numerous life tests with hardened screws under various load conditions, failures have always been the result of a broken ball. The impact life lines in Fig 1 terminate at the loads which will subject the raceways to a mean stress of 550,000 psi. This is considered to be the maximum static, non-Brinell condition for raceways. Tests have shown that ball-bearing screw assemblies can operate for approximately 44,000 impacts at these loads.

When the operating load changes at a constant rate throughout the stroke, the equivalent constant load can be calculated by taking the root mean cube average of the loads:

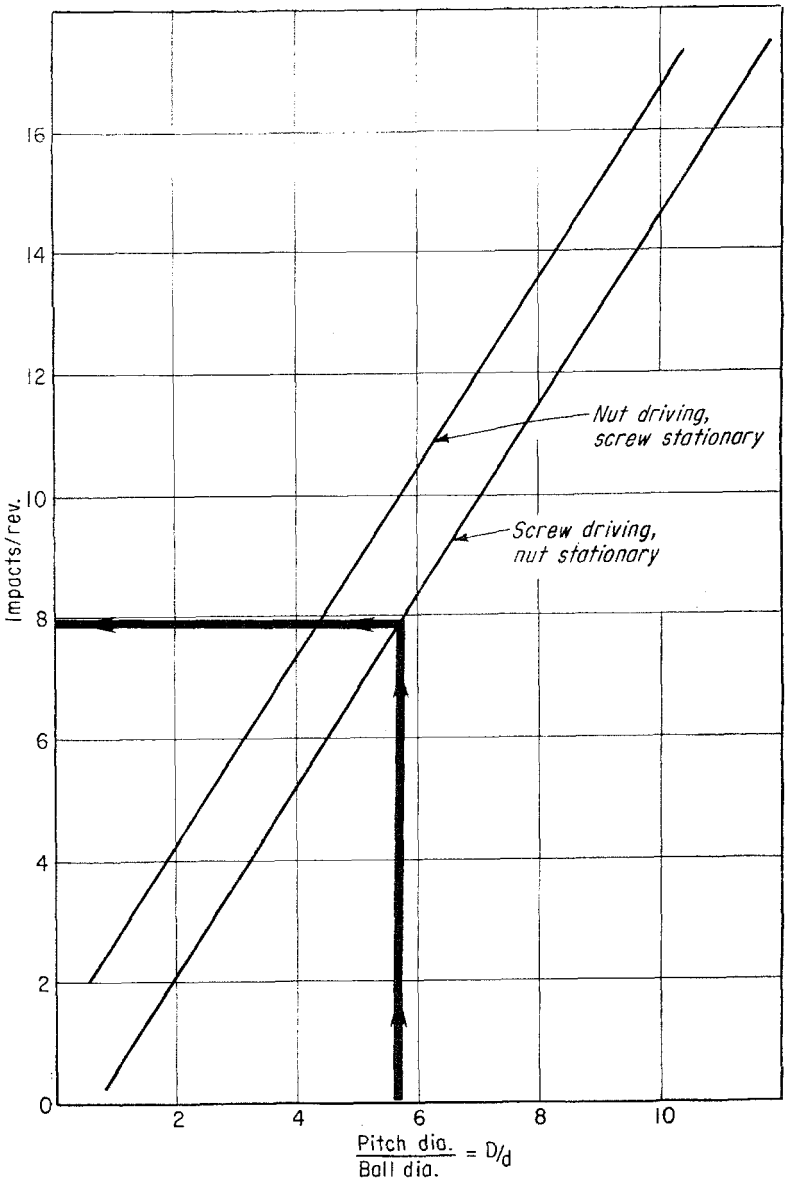
$$L = L_1 + \left(\frac{L_2^3 - L_1^3}{2}\right)^{1/3}$$

where L = the equivalent constant load,
L_2 = the higher load
L_1 = the lesser load

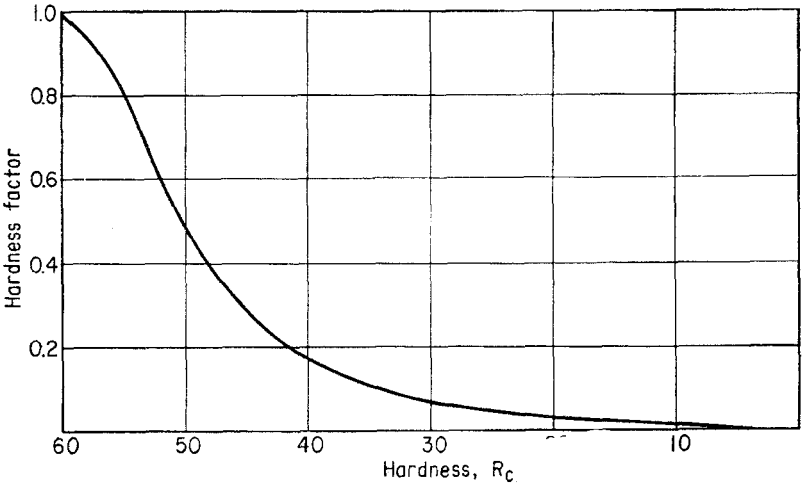
Effect of hardness on life

The life-load chart, Fig 1, is based on a minimum raceway hardness of $60R_c$ and a case depth sufficient to support the load throughout the life of the assembly without appreciable spalling. However, it is sometimes impractical or uneconomical to provide such a degree of hardness.

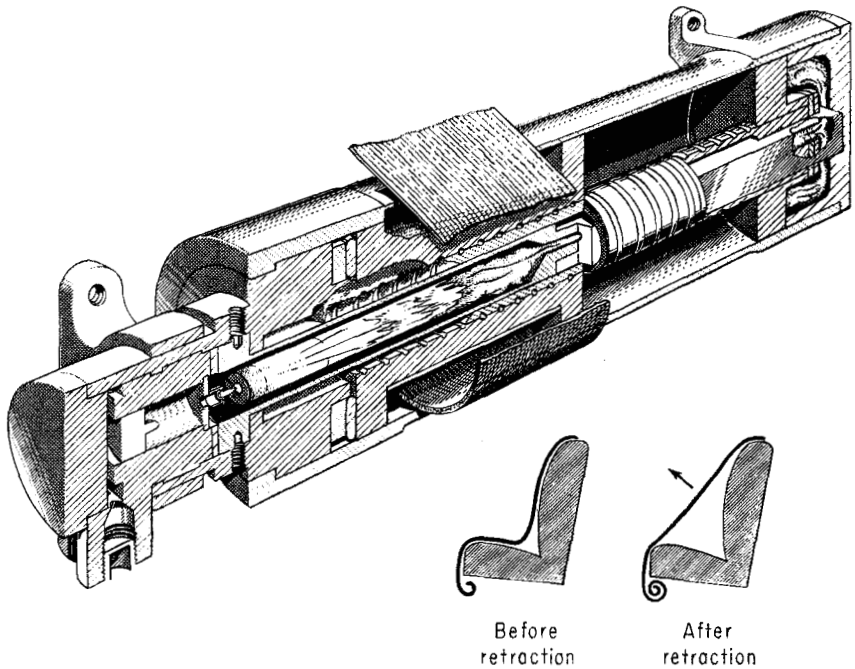
While it is possible to harden very long screws, they will invariably distort as the result of quenching. Straightening of such screws to the required accuracy is difficult and expensive. Hence, a lesser degree of hardness is best for such cases. Also, screws made of stainless steel, such as Armco 17-4PH, are best hardened to between 40 to $45R_c$ by heating to 950 F for 1 hour. This low-temperature heat treatment causes only a minimum of distortion. For lightly loaded, low-cost applications you can



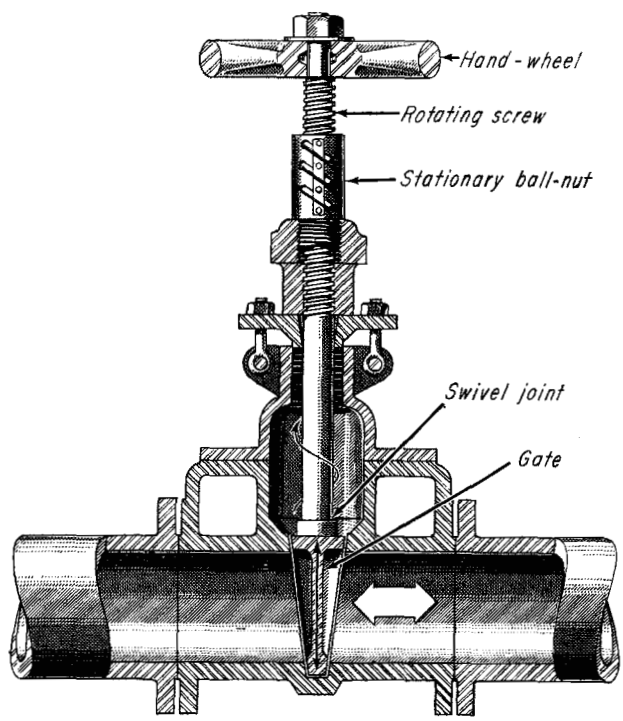
2 • • IMPACTS per revolution versus ratio D/d.



3 • • HARDNESS FACTOR versus Rockwell hardness.



1 Cartridge-operated rotary actuator quickly retracts webbing to forcibly separate a pilot from his seat as the seat is ejected in emergencies. Tendency of pilot and seat to tumble together after ejection prevented opening of chute. Gas pressure from ejection device fires the cartridge in the actuator to force ball-bearing screw to move axially. Linear motion of screw is translated into rotary motion of ball nut. This rapidly rolls up the webbing (stretching it as shown) which snaps the pilot out of his seat. *Talley Industries.*



2 Speedy, easily operated, but more accurate control of flow through valve obtained by rotary motion of screw in stationary ball nut. Screw produces linear movement of gate. The swivel joint eliminates rotary motion between screw and gate.

request cold-rolled unheat-treated steel. However, the hardness for such steel is only approximately 27 to 32R_C.

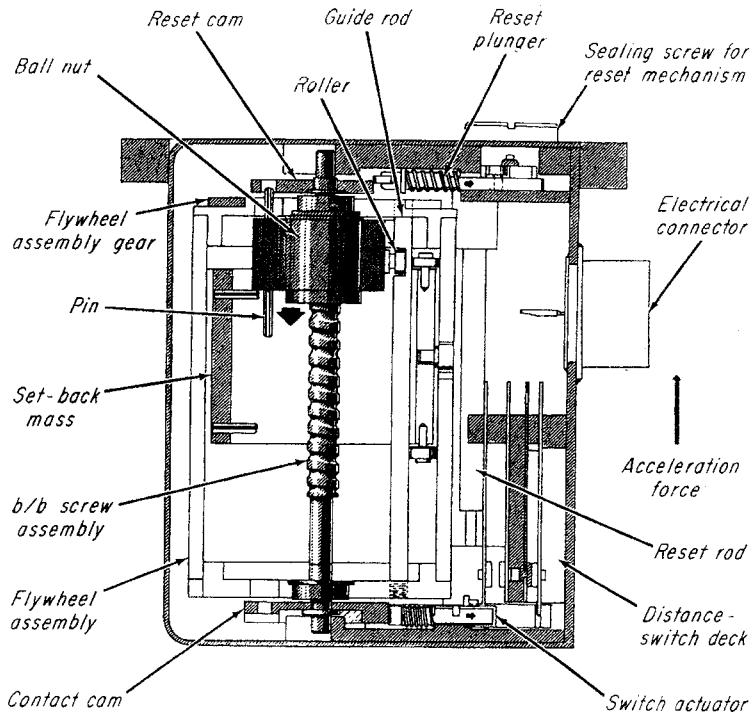
Effect of hardness on the life of ball-bearing screws is shown in Fig 3. Effective load, for determining the life of assemblies, is

$$\text{effective load} = \frac{\text{actual load}}{\text{hardness factor}}$$

Effect of materials on life

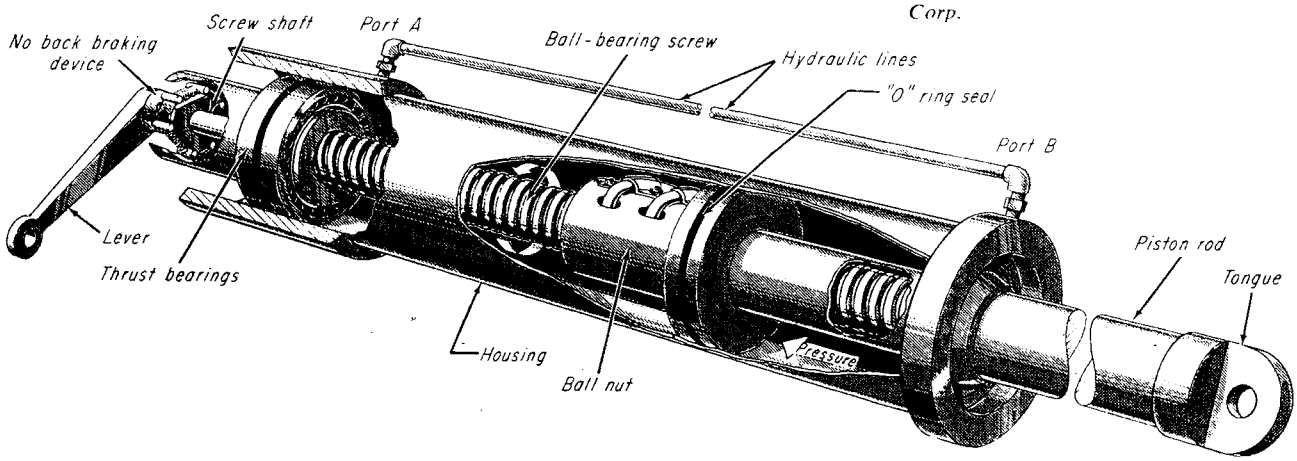
The material employed, if properly hardened and compatible, has little effect on the life of a unit. Most ball-bearing screw assemblies produced by Saginaw are made from SAE 4145, 4150, or 6150 steels, that are usually hardened to 60 R_C.

In the chemical and food-processing industries, actuators are generally



3 Time-delay switching device integrates time function with missile's linear travel. Purpose is to safely arm the warhead. A strict "minimum G-time" system may arm a slow missile too soon for adequate protection of own forces; a fast missile may arrive before warhead is fused. Weight of nut, plus inertia under acceleration will rotate the ball-bearing screw which has a fly wheel on the end. Screw pitch is such that a given number of revolutions of flywheel represents distance traveled. *Globe Industries.*

4 Accurate control of piston position in hydraulic actuator for aircraft has ball-bearing screw mounted directly to piston by means of threaded nut. Piston rod is actuated linearly by means of hydraulic pressure applied to ball nut through port A or B. Linear movement produces rotary motion in screw which is attached to no-back braking device. Piston rod, therefore, can be stopped by any linear position by actuating the lever of braking device. Attaching gear train and rotary dial to screw shaft will give direct reading of linear position of piston rod. *Allison Div of General Motors Corp.*

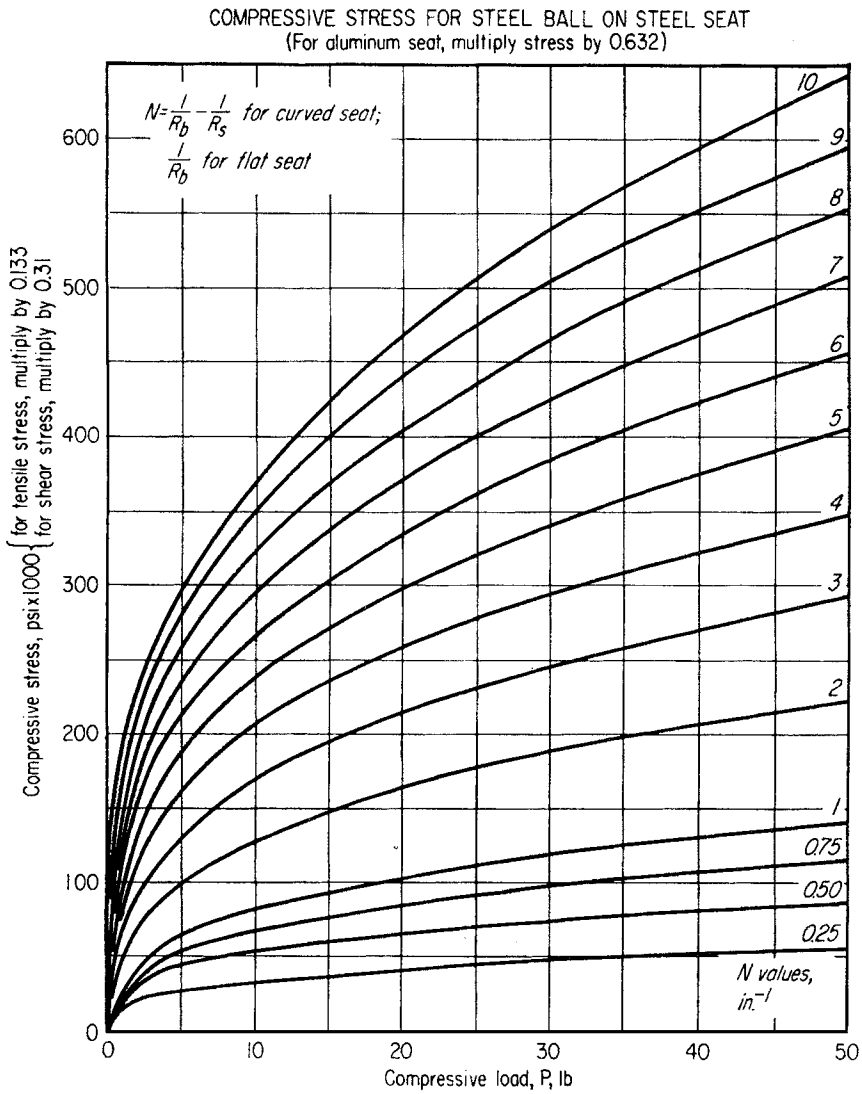


made from corrosion-resistant materials. For high-temperature applications, steels such as the ones listed above are suitable up to about 350 F; AISI Type 440 stainless steel, to 550 F; hot-work tool and die steels, to 800 F; and cobalt-base materials such as Haynes Stellite #25, to 1000 F. The higher temperatures, however, do lower the life of a unit.

Stress on a Bearing Ball

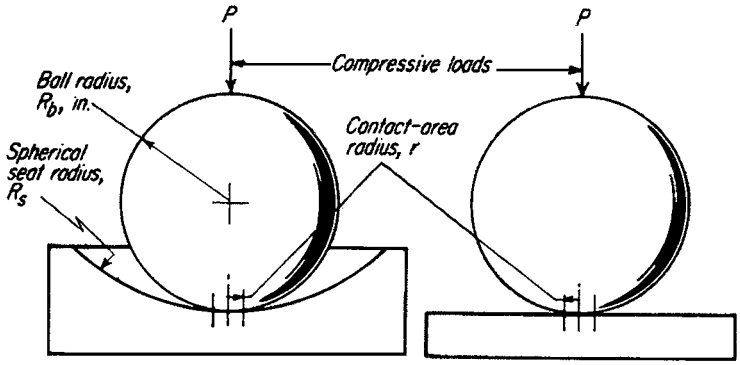
These curves indicate permissible loads when seat is spherical or flat, steel or aluminum.

Jerome E. Ruzicka

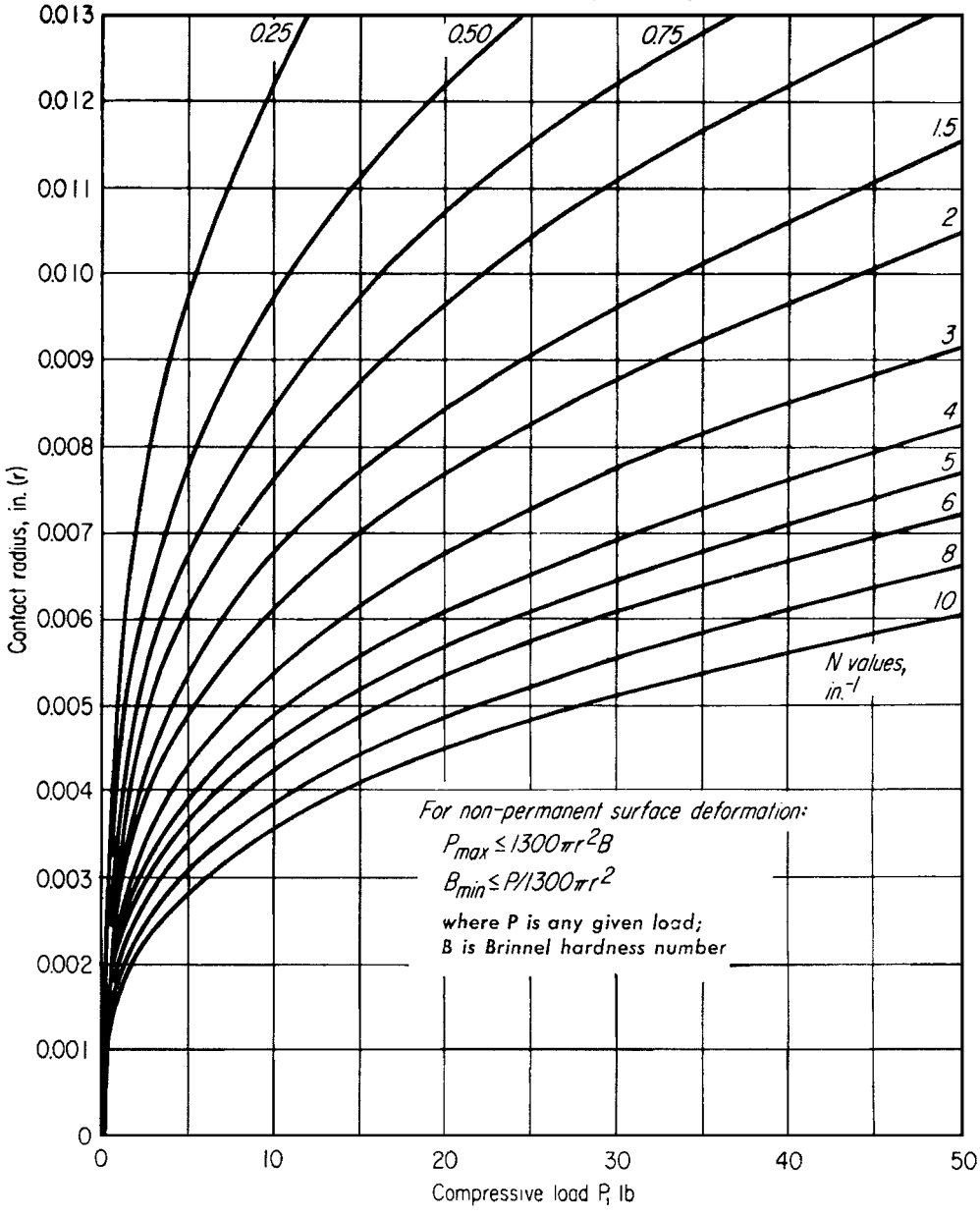


When a design uses steel bearing balls to support a load, it is important to know what stresses result. They are charted on this page. On the continuing page is a diagram that identifies symbols—for applications where the seat is spherical or flat—and also a chart that will help calculate maximum permissible loads.

Symbols used with curves



CONTACT RADIUS FOR STEEL BALL ON STEEL SEAT
(For aluminum seat, multiply radius by 1.25)



Compute Effects of Preloaded Bearings

The trend is to preload whenever precision and spindle rigidity are desired—and tests now show that preload can increase bearing life.

T. A. Harris

MANY designers are failing to take full advantage of the principles of bearing preload. In fact, there are still some misconceptions in this area. Let us immediately clear up two important points:

- Preloading does *not* necessarily decrease bearing life—in fact, bearings which have been preloaded can definitely last longer, provided they are not over preloaded. Design curves included in this article pinpoint the maximum preload for maximum life.
- Preloading need not be restricted to *ball bearings*. Almost any type of rolling-element bearing assembly — ball, cylindrical, roller, tapered roller, spherical—can gain by preloading.

What is preloading?

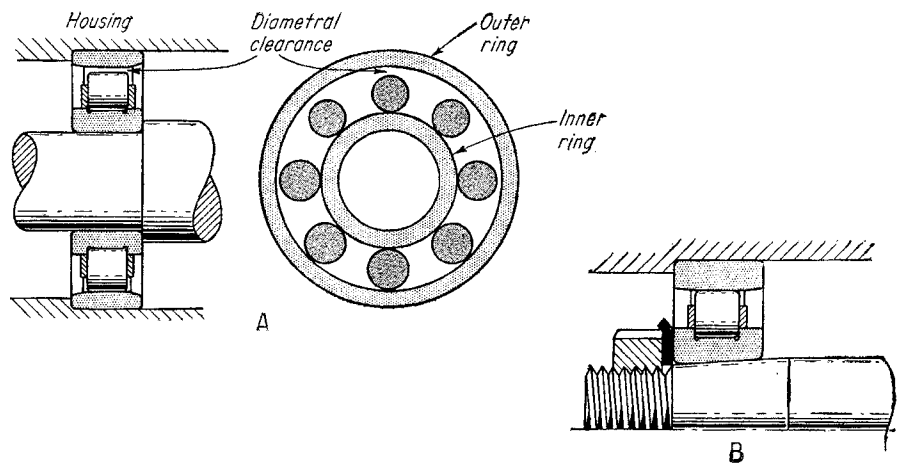
Simply stated, preloading is an internal load applied to a bearing assembly while the assembly is in a stationary (unloaded) condition. This static load, of course, is an addition to that imposed by the weight of the shaft and other members of the assembly.

There are two types of preloading —radial and axial:

Radial preloading can be achieved by any one of these methods:

1. Forcing the bearing on a tapered shaft to expand the inner ring and thus take up the clearance (Fig 1A to 1B).
2. Mounting a tapered sleeve on a shaft, and mounting the bearing on to the sleeve (C).
3. Employing an elliptically out-of-round outer ring (D). The ring snaps over a normal inner ring-and-roller set of a cylindrical roller bearing. This provides a selective radial preload in the vicinity of the minor axis and prevents skidding in high-speed, lightly loaded applications.
4. Tightening tapered roller bearings against each other (Fig E and F). Although this is an axial tightening, in effect radial preloading is achieved which improves roller load distribution and increases life.
5. Selecting more interference in the fit between the outer diameter of the bearing and its housing.
6. Heating the outer ring and allowing it to shrink over the rollers after assembly. A line-to-line fit is then required on the shaft and the housing.

Axial preloading, Fig 2, is usually obtained by shifting the inner or outer race of the bearing in the axial direction. This can be done in several ways: by grinding the side of one ring so that it is normally at an offset position in comparison to the other ring,



1 A. Diametral (radial) clearance found in most off-the-shelf rolling-element bearings. One object of preloading is to remove this clearance during assembly.
B. Cylindrical roller bearing mounted on tapered shaft to expand inner ring. Such bearings are usually made with a taper on the inner surface of 0.04 in./in.
C. Spherically roller bearing mounted on tapered sleeve to expand the inner ring.
D. Elliptically out-of-round outer ring is snapped over the normally round inner ring of a cylindrical-roller bearing to prevent skidding during high-speed operation.
E. Tapered roller bearings are locked together axially to obtain radial preload.
F. Lightly preloaded tapered roller bearings in the wheel of a mine car.

and then by mounting the bearing in pairs (A to D); by use of shims (E); and by the insertion of spacers in which one spacer is slightly longer than the other (F).

What does preloading do?

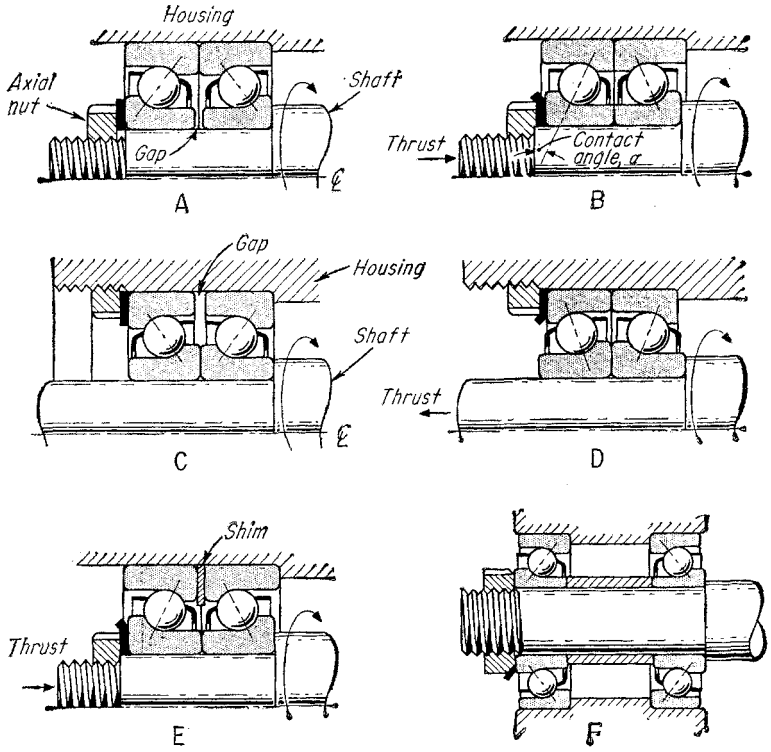
Preloading removes the internal clearances that normally exist between the balls (or rollers) and one of the races. In fact, because the result is usually an interference fit between the balls and the races, clearance or play is avoided even under load (up to, of course, a specific point). Thus, preloading:

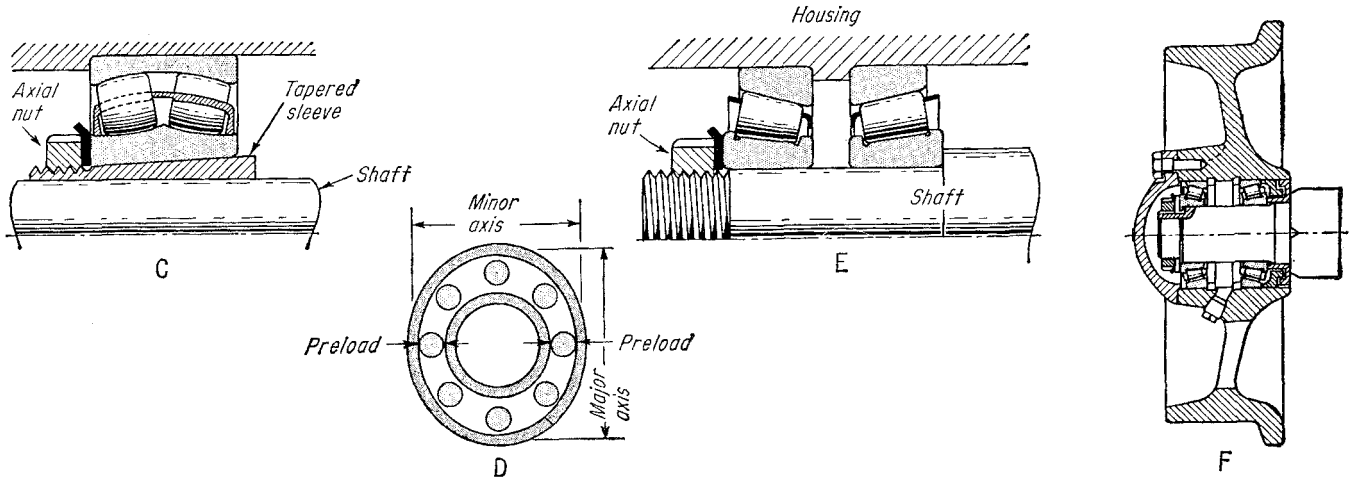
- **Provides more accurate shaft positioning,** both axially and radially. This is a prime objective for designers of precision tools and mechanisms, such as machine tool spindles, instruments, gyroscopes. Of course, many designers in these fields are already employing preload.
- **Reduces the shaft deflection under load** and improves the assembly stiffness characteristics.
- **Increases the bearing fatigue life,** providing that the assembly is not overpreloaded.
- **Decreases bearing noise** and permits the bearing to take higher shock.
- **Provides system isoelasticity,** in which the deflection in the bearing system is along the line of the external load.

Care must always be taken to avoid excessive preload because this increases the running torque and operating temperature of the bearing and thus significantly reduces bearing life. The following sections give the key equations and charts for accurately predicting the amount of preload a bearing assembly should have. Sample problems are included in most cases.

continued, page 86

2 A. Duplex set with back-to-back angular ball bearings prior to axial preloading. The inner ring faces are ground to provide a specific gap.
B. Same unit as in (A) after tightening axial nut to remove gap. The contact angles will have increased.
C. Face-to-face angular-contact duplex set prior to preloading. In this case it is the outer-ring faces which are ground to provide the required gap.
D. Same set as in (C) after tightening the axial nut. The convergent contact angles increase under preloading.
E. Shim between two standard-width bearings avoids need for grinding the faces of the outer rings.
F. Precision spacers between bearings automatically provide proper preload by making the inner spacer slightly shorter than the outer.





RADIAL PRELOADING

Preload vs bearing life

As stated previously, light preloading increases the bearing fatigue life. Specifically, in the case of radial preloading, the preload extends the circumferential arc of loading (Fig 3), which in turn reduces the maximum load experienced by a ball or roller.

But by how much is the bearing life extended? Most statements on preload are qualitative; quantitative analyses are generally shunned as being too complicated. This was perhaps true in the past. Now, with certain key equations and charts, one can directly come up with accurate estimates as to the amount of preload that is desirable and its effect on bearing life.

First step is to determine the extent of the circumferential zone of rolling element loading. This is obtained by solving Eq 1 and 2 simultaneously for δ, the radial deflection, and ϵ, the projection of the zone of loading on the bearing pitch diameter of symmetry (a numerical problem that follows illustrates the technique):

$$F = ZK\left(\delta - \frac{c}{2}\right)^n J \qquad (1)$$

$$\epsilon = \tfrac{1}{2}\left(1 - \frac{c}{2\delta}\right) \qquad (2)$$

where F is the applied load on the bearing (caused by the load imposed on the shaft from the gearing, belting, rotating mass, etc), Z is the number of balls or rollers, K is the deflection constant defined for most deep-groove ball bearings by Eq 3 and for roller bearings by Eq 4, c is diametral clearance (which is frequently referred to as radial clearance according to Anti-Friction Bearing Manufacturers' Association (AFBMA) terminology), and J is a radial load function given by Fig 4 for ball and roller bearings. The exponent n is 1.5 for ball bearings and 1.1 for roller bearings. For ball bearings

$$K = 1.53 \times 10^7 D^{0.5} \qquad (3)$$

For roller bearings

$$K = 5.28 \times 10^6 L_r^{0.89} \qquad (4)$$

where D is the diameter of the balls and L_r the effective length of the rollers.

You can easily solve Eq 1 and 2 by trial-and-error techniques. Assume a value of ϵ, then pick off J in Fig 4. Next, solve for δ in Eq 1 and use this value in Eq 2 to determine a new value of ϵ, which you then compare against the assumed value. Repeat the process until the difference between the assumed and the calculated values of ϵ is sufficiently small (usually under 0.01).

This value of ϵ will then affect the rating life or L_{10} fatigue life, which is in terms of hours of a radially loaded, rolling bearing in accordance with AFBMA load rating standards given by the equations:

For ball bearings

$$L_{10} = \frac{10^6 \lambda}{60N}\left[\frac{C}{F}\right]^3 \qquad (5)$$

For roller bearings

$$L_{10} = \frac{10^6 \lambda}{60N}\left[\frac{C}{F}\right]^{10/3} \qquad (6)$$

In the above equations, C is the basic load rating supplied by the bearing catalog, and N the shaft speed. **These equations, however, differ from the often published AFBMA equations in that they contain a life adjustment factor** λ. This factor is obtained from Fig 5 by knowing ϵ, and thus accounts in Eq 5 and 6 for the effect of diametral clearance, both positive and negative, on bearing life.

Generally, in nonpreloaded bearings, the clearances are relatively large and the values for λ quite low, in the 0.7 to 0.9 range (hence it is frequently called a "reduction factor"). But with preloaded bearings, values above 1.0 are readily obtained. In addition, values of ϵ greater than 1 should be avoided to maintain long fatigue life. Good design practice calls for radial preloads which cause ϵ to fall between 0.5 and 1.0. Improved fatigue life is thereby obtained.

Example I—Nonpreloaded life

A single-row deep-groove ball bearing (SKF bearing number 6309 with a loose C3 fit) has a basic dynamic load rating of 9120 lb. This bearing supports a radial load of 2000 lb at a shaft speed of 1000 rpm. According to the catalog, the bearing contains 8 balls of $\frac{11}{16}$ in. diameter. Also, this bearing is listed as having a mean diametral clearance of $c = 0.001$ in. Without any preload, what is the radial deflec-

Symbols

Symbols	Description	Units	Source
B	total groove curvature	—	Eq 14
c	diametral (radial) clearance	in.	bearing mfr or AFBMA tables
C	basic load rating	lb	catalog
d	bearing pitch diameter	in.	catalog
D	ball or roller diameter	in.	catalog
f_i	inner ring groove radius/D	—	bearing mfr.
f_o	outer ring groove radius/D	—	bearing mfr.
F	radial load or preload	lb	bearing application
F_1	axial load on bearing 1	lb	Eq 13 and 15
F_2	axial load on bearing 2	lb	Eq 13 and 15
G	axial deflection constant	—	Fig 9
J	radial distribution integral	—	Fig 4
K	radial deflection constant	—	Eq 3 or 4
L_{10}	rating life (10% failures)	hr	Eq 5 or 6
L_r	effective roller length	in.	catalog
N	shaft speed	rpm	bearing application
T	external thrust load	lb	bearing application
Z	number of balls or rollers	—	catalog
α_0	zero load contact angle	deg	bearing mfr.
α_1	contact angle on bearing 1	deg	Eq 20 and 21
α_2	contact angle on bearing 2	deg	Eq 20 and 21
δ	radial or axial deflection	in.	Radial: Eq 1 and 2 Axial:
δ_p	axial preload deflection	in.	Fig 10
Δ	increase in clearance due to centrifugal force	in.	Eq 11 or 12
ϵ	projection of loading arc on bearing diameter	—	Eq 2
λ	life adjustment factor	—	Fig 5

Note: When source is listed as "bearing mfr.," the data may be found in catalogs.

tion and estimated L_{10} fatigue life?
From Eq 3

$$K = (1.53(10^7)(0.6875)^{0.5} = 1.269 \times 10^7$$

From Eq 1

$$2000 = (8)(1.269)(10^7) \times \left(\delta - \frac{0.001}{2}\right)^{1.5} J$$

$$0.0000197 = (\delta - 0.0005)^{1.5} J \quad (7)$$

From Eq 2

$$\epsilon = 0.5\left(1 - \frac{0.001}{2\delta}\right) = 0.5\left(1 - \frac{0.0005}{\delta}\right) \quad (8)$$

Assume a value for ϵ (a good starting point for nonpreloaded bearings is $\epsilon = 0.4$). Use the ϵ value in Fig 4 to determine J, solve for δ in Eq 7, and then solve for ϵ in Eq 8 to see how close it is to the assumed value. This finally yields:

$$\epsilon = 0.402$$
$$\delta = 0.00254 \text{ in.}$$

Now compute the predicted bearing life. At $\epsilon = 0.402$, from Fig 5, $\lambda = 0.9$ and L_{10} from Eq 5 becomes:

$$L_{10} = \frac{(1{,}000{,}000)(0.9)}{(60)(1000)}\left[\frac{9120}{2000}\right]^3 = 1438 \text{ hr}$$

Example II—Preloaded life

Let us now look at what happens when the bearing of the foregoing example (bearing No. 6309) is mounted with a press fit on the shaft and in the housing such that the resultant clearance is 0.0005 in. tight. This provides a light radial preload. What radial deflection and L_{10} fatigue life can now be expected?
From Eq 1

$$2000 = (8)(1.269)(10^7) \left(\delta + \frac{0.0005}{2}\right)^{1.5} J$$

$$0.0000197 = (\delta + 0.00025)^{1.5} J \quad (9)$$

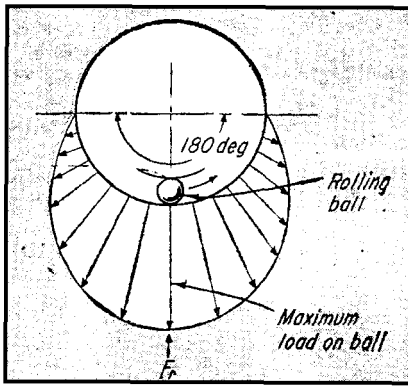
From Eq 2

$$\epsilon = 0.5\left(1 + \frac{0.0005}{2\delta}\right) = 0.5\left(1 + \frac{0.00025}{\delta}\right) \quad (10)$$

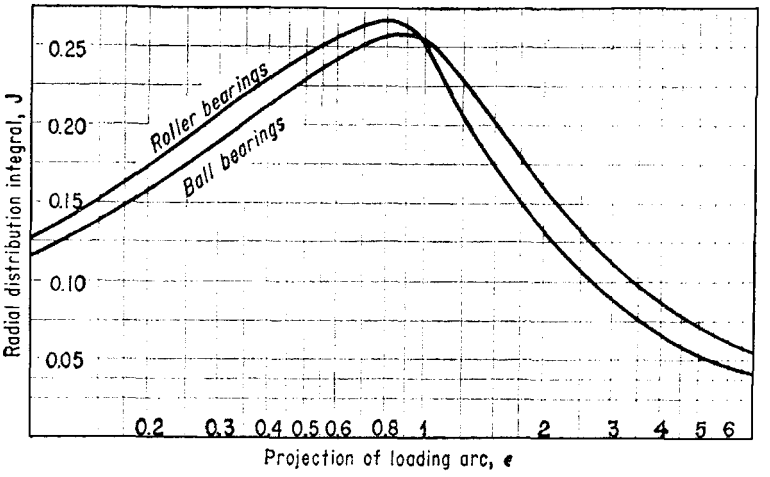
Solving Eq 9 and 10 with the aid of Fig 4 yields:

$$\epsilon = 0.577$$
$$\delta = 0.0016 \text{ in.}$$

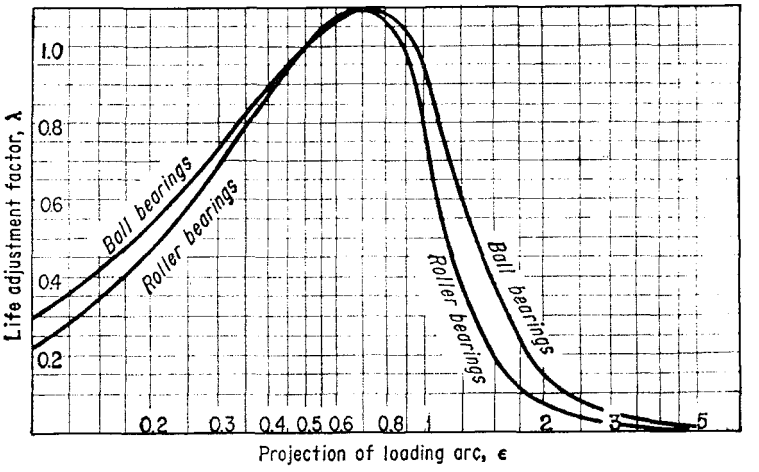
At $\epsilon = 0.577$, from Fig 5, $\lambda = 1.055$. Thus from Eq 5:



3 Distribution of load among the rolling elements of a radially loaded ball or roller bearing having no diametral clearance (with radial preload). Note that the arc of loading is 180 deg. An unpreloaded bearing would have a much smaller loading arc, fewer of the balls or rollers will be taking up the radial load, and hence the maximum load will be higher (than in a preloaded bearing) which decreases life.



4 Radial load function J vs the zone of-loading projection, ϵ. These factors play an important part in computing the change in fatigue life of a ball bearing when preloaded. For best results, design for $0.5 < \epsilon < 1$.



5 Fatigue-life reduction factor, λ vs ϵ. Many designers are unaware that this factor should be applied to the standard fatigue equations used in industry to predict the life of roller bearings. To obtain values of 1.0 or over for λ (desirable), factor ϵ should be between 0.5 to 0.9.

$$L_{10} = \frac{(1{,}000{,}000)(1.055)}{(60)(1000)}\left(\frac{9120}{2000}\right)^3$$

$$= 1660 \text{ hr}$$

Hence, this bearing when mounted with 0.0005-in. interference deflects 0.0009 in. less and has a 15% increase in fatigue life.

Equations for high speeds

The previous analysis did not take into consideration the centrifugal force associated with the ball or roller orbital speed. At high cage speeds, the centrifugal force tends to increase the diametral clearance which reduces bearing life. Because this force increases as the square of cage speed, its effect at slow speeds is usually neglected.

The increase in clearance, Δ, caused by centrifugal force can be approximated by the following equations. This calculated value should be added to the value for c used in Eq 1 and 2: For ball bearings

$$\Delta = 1.07 \times 10^{-9} D^{1.67} d^{0.67} N^{1.33} \times \left[1 \pm \frac{D}{d}\right]^{1.33} \quad (11)$$

For roller bearings

$$\Delta = 2.38 \times 10^{-12} D^{1.8} L_r^{0.1} d^{0.9} N^{1.8} \times \left[1 \pm \frac{D}{d}\right]^{1.8} \quad (12)$$

In these equations d is the bearing pitch diameter. Use the minus sign when the inner ring is rotating (as when the shaft is rotating) and the plus sign for outer ring rotation.

For the bearing in Example II (pitch dia = 2.8543 in.), rotating at only 1000 rpm, the increase in clearance from Eq 11 is 0.000008 in.—negligible when compared to the 0.0005 diametral tightness. On the other hand, if the shaft speed is raised to 10,000 rpm, the estimated increase in clearance will be 0.0002 in. which must be subtracted from the preload tightness. Roughly, this will result in a λ factor of 1.03, which will decrease life by about 2.5%. Thus, when designing for a radially preloaded bearing application at other than slow speeds it is necessary to account for the effect of bearing rotational speed.

The clearance in Eq 1 and 2 is that which occurs *after* the bearing is mounted on the shaft and in the housing. When the shaft or housing is other than steel (assuming steel bearings), the effect on clearance of differential expansion due to elevated operating temperatures must be taken.

AXIAL PRELOADING

The most common type of axial preloading occurs in angular contact ball-bearing applications in which duplex bearings are pressed axially against each other to gain increased rigidity against the effect of externally applied thrust load.

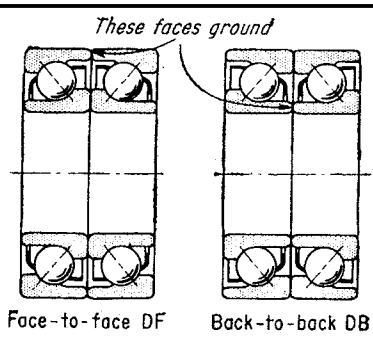
The reason for the improvement in rigidity, or *stiffness*, is illustrated by Fig 7 which shows ball-bearing deflection as a function of bearing load. If the bearing can be made to operate to the right of the knee of the curve, ie, if the left-hand portion of the curve could be removed, the subsequent deflection with load can be decreased considerably because the deflection rate diminishes as load increases.

For roller bearings, however, the deflection-load characteristic is nearly linear and there exists no knee to be removed. Consequently roller bearings are rarely preloaded to increase stiffness. Tapered roller bearings, however, require an axial load for proper operation, and in the absence of an applied thrust load this may be effected by applying a light axial preload.

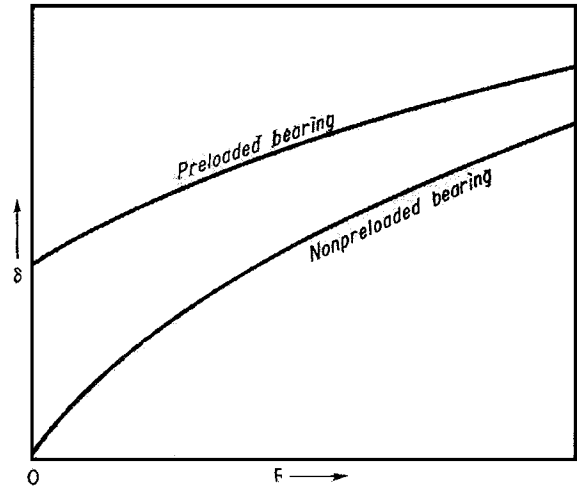
Angular contact ball bearings can be purchased from manufacturers' catalogs to yield specified preloads. The bearings usually have one side face ground down. When such bearings are duplex mounted and locked up against each other as in Fig 6, a specified preload exists according to the difference in width between the inner and outer rings.

For example, SKF angular contact ball bearings which carry the suffix G followed by a code symbol indicate the amount of preload; thus G02 indicates 20 lb and G2 indicates 200 lb preload. Table I (opposite) gives a schedule of preloads supplied by SKF.

However, if you wish to use stand-



6 Duplex sets of angular contact ball bearings. The back-to-back is the more popular arrangement because the contact angle converges outside the bearing outer ring which provides a high degree of resistance to misaligning forces. Select these bearings when loading is cantilevered or overhung as for pulleys, sheaves. Face-to-face mountings are best when it is desired to dismount spindles and other accessories that are clamped against the inner ring of the bearing—but without relieving the preload of the bearings.



7 Deflection vs load characteristics for ball bearings. As the load increases, the rate of the increase of deflection is slowed, therefore preloading (top line) tends to reduce the bearing deflection under additional loading.

ard bearings, you can use a shim of a width to match the amount that is normally ground off from a preload bearing. Because this amount for a specific preload varies with the bearing type, you must compute this value (see the technique that follows), or you may be able to obtain specific values from the bearing manufacturers.

Computing the grindoff amount

Angular-contact ball bearings that are to be preloaded are usually mounted in pairs in a face-to-face or back-to-back mounting. This mounting may be subjected to an additional, applied thrust load, T. The equilibrium of axial forces requires that

$$T = F_1 - F_2 \qquad (13)$$

where F_1 and F_2 are the thrust loads on bearings 1 and 2, Fig 8. If there is only preload on the bearings (no applied thrust load) then

$$F_1 = F_2.$$

The next important relationship involves the inner and outer raceway groove curvatures, f_i and f_o, which can be obtained from the bearing manufacturers. A constant, B, is then obtained by means of the equation

$$B = f_i + f_o - 1 \qquad (14)$$

The groove curvatures are usually given as a percentage of the ball diameter and fall between 52 to 53% of the ball diameter for most angular-contact bearings.

We now employ two equations to relate the axial deflection, δ, to the axial preload, F:

$$F_j = ZD^2G \times \sin\alpha \left[\frac{\cos\alpha_0}{\cos\alpha} - 1\right]^{1.5} \qquad (15)$$

$$\delta_j = BD\frac{\sin(\alpha - \alpha_0)}{\cos\alpha} \qquad (16)$$

Here, subscript j relates to the specific bearing in question either bearing No. 1 or 2 in a duplex set, α_0 is the initial contact angle (under zero load conditions) and α is the final contact angle.

Values for Z, D, and α_0 in the above equations are easily obtained from catalogs or from the bearing manufacturers. The axial preload, F, is usually known or assumed from the application requirements.

Go to the curve in Fig 9 to obtain a value for G based on the computed value for B (from Eq 14), and to the chart in Fig 10 to obtain other necessary factors as follows:

1. Calculate a constant, t, from the known factors in the first part of Eq 15, by making t equal to

$$t = \frac{F}{ZD^2G} \qquad (17)$$

2. In Fig 10, locate the point of intersection of the line for t and the radial line for α_0. On the curves, the example is $t = 0.01$ and $\alpha_0 = 40$ deg.
3. Swing a radius about the right-hand origin through the located point.
4. At the intersection of this arc and the abscissa line (where $\alpha_0 = 0$) locate the value of δ/BD. In the example $\delta/BD = 0.089$.
5. Align a straight-edge through the intersection of t and α_0 lines such that the straight-edge is parallel to identically numbered markers of the upper and lower $\alpha - \alpha_0$ scales. In the example, locate $\alpha - \alpha_0 = 3.6$ deg.

From the values obtained in steps 4 and 5, you can now quickly determine the axial deflection δ and final contact angle α—without need for further reference to Eq 15 and 16. The amount of grinding required to achieve a given preload is then equal to δ.

Example III—Axial preload on duplex pair

It is desired to obtain an axial preload of 500 lb from a set of duplex angular contact ball bearings. The bearings have 52% inner and outer raceway groove curvatures, an initial contact angle of 40 deg, and a complement of 15 balls of 0.5 in. diameter. How much stock must be ground from the inner ring face of each bearing? From Eq 14:

$$B = 0.52 + 0.52 - 1 = 0.04$$

From Fig 9, for a value of $B = 0.04$, $G = 110{,}000$.

From Eq 17:

$$t = \frac{500}{15 \times (0.5)^2 \times 110{,}000} = 0.0012$$

From Fig 10, $\delta/BD = 0.022$. Hence,

$$\delta_p = (0.022)(0.04)(0.5) = 0.00044 \text{ in.}$$

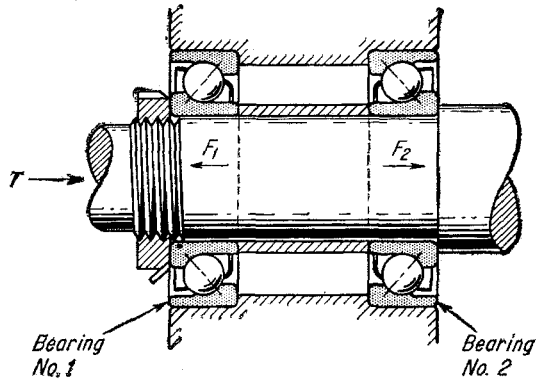
Subscript p was added to denote that the deflection is due to axial preloading alone.

SKF preload suffixes for bearings

Table I

Bore dia. mm		Light preload		Heavy preload	
Over	Incl.	Lb	Suffix	Lb	Suffix
0	20	20	G 02	100	G 1
20	45	50	G 05	200	G 2
45	80	100	G 1	300	G 3
80	95	100	G 1	400	G 4
95	120	200	G 2	500	G 5
120	150	200	G 2	700	G 7
150	240	300	G 3	900	G 9

Bearings can be ordered with faces of inner ring shaved down to provide a specific preload. Example: To obtain a heavy preload for a 7210 B angular contact bearing, specify 7210 BG 5. This bearing will provide a 500-lb axial preload when clamped in assembly.



8 Preloaded set of duplex bearings subjected to an external thrust load, T. The computation for the resulting deflection is complicated by the fact that the preload at bearing 1 is increased by load T, while the preload at bearing 2 is decreased.

Also from Fig 10, $\alpha - \alpha_0 = 0.9$ deg. Hence,

$$\alpha = 0.9 + 40 = 40.9 \text{ deg}$$

Thus, 0.00044 in. must be removed from the inner ring face of each bearing to obtain 500 lb preload.

Example IV—With external thrust loads

Suppose now that the preloaded duplex bearings of Example III are subjected to external thrust load and it is necessary to obtain the axial deflection (Fig 8). **This complicates the analysis because only one of the two bearings resists this thrust.**

Designating the axial deflection of the bearing set due to the thrust load alone as δ_t, then at bearing 1, the effective axial deflection is $\delta_t + \delta_p$, in which δ_p is the axial deflection due to preloading. Correspondingly, the axial deflection at bearing 2 is $\delta_p - \delta_t$. The latter condition exists as long as bearing 2 is not relieved of all load. Therefore, according to Eq 16:

$$\delta_p + \delta_t = BD \frac{\sin(\alpha_1 - \alpha_0)}{\cos \alpha_1} \quad (18)$$

$$\delta_p - \delta_t = BD \frac{\sin(\alpha_2 - \alpha_0)}{\cos \alpha_2} \quad (19)$$

These two equations can be added together to yield:

$$2\delta_p = BD \left[\frac{\sin(\alpha_1 - \alpha_0)}{\cos \alpha_1} + \frac{\sin(\alpha_2 - \alpha_0)}{\cos \alpha_2} \right] \quad (20)$$

Also, according to Eq 14 and 15:

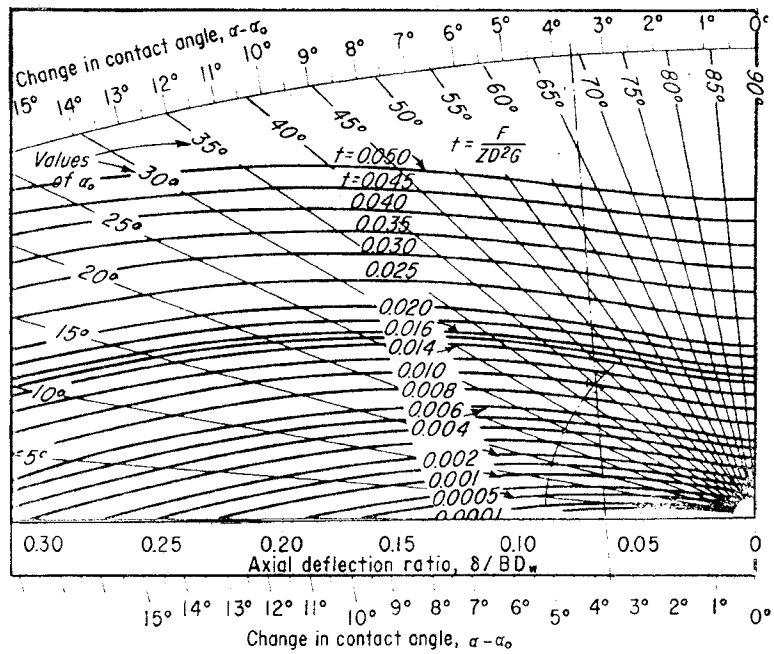
$$T = ZD^2G \left\{ \sin \alpha_1 \left[\frac{\cos \alpha_0}{\cos \alpha_1} - 1 \right]^{1.5} - \sin \alpha_2 \left[\frac{\cos \alpha_0}{\cos \alpha_2} - 1 \right]^{1.5} \right\} \quad (21)$$

Eq 19 and 20 must be solved simultaneously for the unknown contact angles α_1 and α_2. This procedure is best done on a digital computer; **however, several graphical techniques can be employed in the following manner:**

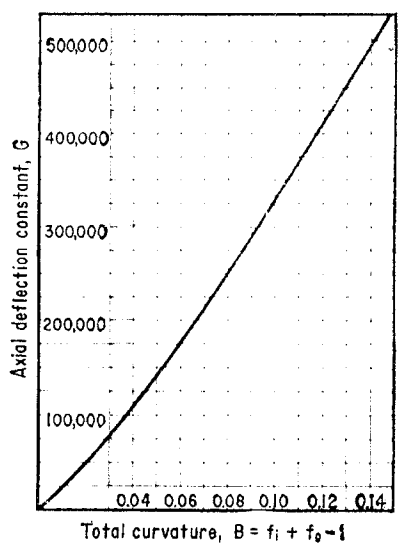
Example V—Simplified graphical techniques

Determine the axial deflection caused by an external 1000-lb thrust load applied to the preloaded duplex bearing set of Example III.

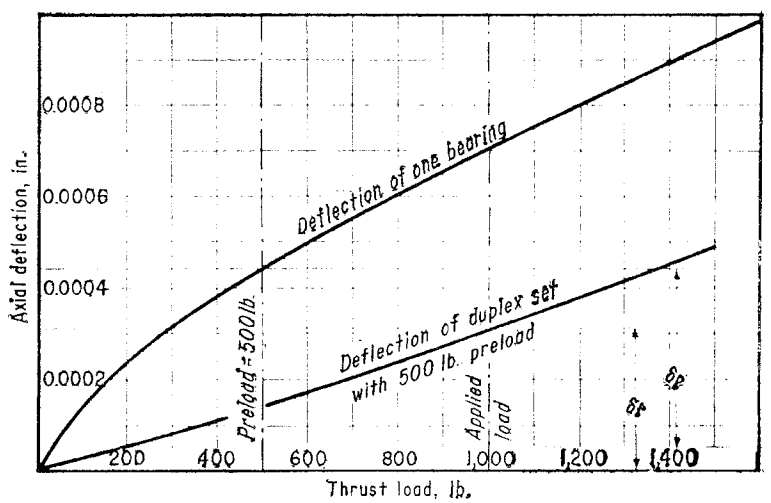
Establish for each bearing its load-deflection curve caused by preloading alone. This is done by selecting a schedule of loads (from 200 to 1600 lb, Table II), and then by Fig 10 and the method in Example III a series of deflections, δ_j, are obtained.



10 **Design chart for computing the axial deflection, δ, under load, and the resultant change in contact angle α. Dashed lines are for Ex. III.**



9 **Axial-deflection constant, G, as a function of the curvatures of the inner and outer ball grooves, f_i and f_o.**



11 **Axial deflection vs thrust load** for sample calculations. The chart is constructed for one set of conditions, then employed to determine the deflection of duplex sets, such as the bearings illustrated in Fig 8.

Now plot the values of δ_j vs F_j (Fig 11). Note that at the preload of 500 lb, δ_p on each bearing is 0.00044 in. This value checks with that of Example III.

Next, with the aid of Fig 11, compute the effect of additional axial deflections caused by external axial thrusts. Thus in Table III, an additional deflection of 0.0001 results in a total deflection of 0.00054 on bearing 1, and 0.00034 on bearing 2. From Fig 11, the corresponding bearing loads are 670 and 340 lb, respectively. These loads act against each other, hence the algebraic sum is T = 670 − 340 = 330 lb.

Finally, plot T versus δ_t on Fig 11. From the latter curve at 1000 lb applied load, the deflection of the duplex bearing set in the loading direction is 0.0003 in. The thrust load on bearing 1 is 1070 lb and on bearing 2, 70 lb. If a single bearing were used to support the 1000 lb load with no help from preloading, the axial deflection would be 0.0007 in.

An approximate — but simpler — graphical method is also available. Note that to relieve the preload on bearing 2 in Example IV, the axial deflection δ_t be equal to the preload deflection δ_p. Therefore on Fig 11, find at $\delta_1 = 2\delta p = 0.00088$, thrust load $T = 1380$ lb. Now at $T = 1380$ lb and $\delta = 0.00044$ in. describe the point on the load-deflection curve of the duplex bearing set at which bearing 2 becomes unloaded. Draw a straight line on Fig 11 between the origin and the point just determined. This straight line approximates the deflection curve of the duplex bearing set. At $T = 1000$ lb find $\delta_t = 0.00033$, which value is practically identical to that previously determined.

In the foregoing examples, the bearings in the duplex set were identical. This condition was picked to simplify the calculations; however, it is sometimes desirable to use duplex bearings which are different in ball complement, ball size, groove curvature, and contact angle. In that case curves of δ_j versus F_j must be determined for each bearing. Thereafter, the calculated procedure is identical.

Load-deflection calculations for the examples
Table II

F_j	$t=\frac{F_j}{ZD^2G}$	$\frac{\delta_j}{BD}$	$\alpha_j-\alpha_o$	δ_j
200	0.0005	0.012	0.5	0.00024
400	0.0010	0.019	0.8	0.00038
600	0.0015	0.025	1.05	0.0005 0
800	0.0019	0.029	1.2	0.00058
1000	0.0024	0.034	1.4	0.00068
1200	0.0029	0.039	1.6.	0.00078
1400	0.0034	0.044	1.8	0.00088
1600	0.0039	0.049	2.0	0.00098

Table III

δ_t	$\delta_p+\delta_t$	F_1	$\delta_p-\delta_t$	F_2	T
0.0001	0.00054	670	0.00034	340	330
0.0002	0.00064	870	0.00024	200	670
0.0003	0.00074	1070	0.00014	100	970
0.0004	0.00084	1290	0.00004	20	1270

Table IV

δ_t	$\delta_{p_1}+\delta_t$	F_1	$2F_1$	$\delta_{p_2}-\delta_t$	F_2	T
0.0001	0.00038	400	800	0.00034	340	460
0.0002	0.00048	570	1140	0.00024	200	940
0.0003	0.00058	760	1520	0.00014	100	1420

Y factors — effect of preload on bearing life
Table V

Contact angle, deg	20	25	30	35	40
Y-factor	1.00	0.87	0.76	0.66	0.57

Triplex bearings

Frequently, two identical angular contact bearings mounted in tandem are reverse-mounted with a third angular contact bearing and preloaded, Fig 12. In this case, Eq 13 becomes:

$$T = 2F_1 - F_2 \qquad (22)$$

The axial deflections in the bearings are given by the equations below:

$$\delta_1 = \delta_{p1} + \delta_t \qquad (23)$$
$$\delta_2 = \delta_{p2} - \delta_t$$

in which δ_{p1} and δ_{p2} are the preload deflections in each bearing. This arrangement can increase bearing stiffness more than a simple duplex mounting. The following example illustrates this condition.

Example V—Three-bearing design

Three angular-contact ball bearings, identical to those of Example III, are mounted with two in tandem and one opposed (Fig 12). The bearings are preloaded to 500 lb. What axial deflection may be expected from an applied thrust load of 1000 lb?

Table II has already given the deflection load for each bearing (this was plotted on Fig 11). The preload in each bearing 1 is 250 lb each and δ_{p1} is 0.00028 in. As before, δ_{p2} is 0.00044 in. The data from Table III is revised to that given in Table IV. Thus the axial deflection is reduced to 0.0002 in. in lieu of 0.0003 in. as calculated for the duplex bearings set. Accordingly, the load on each of bearing 1 is 590 lb and on bearing 2 is 180 lb.

Changes in fatigue life

Will the fatigue life of duplex bearings be increased by axial preload, as was the case with the radially preloaded bearings? According to the AFBMA standard on load rating of ball bearings, the rating life of a thrust-loaded radial ball bearing is given by:

$$L_{10} = \frac{10^6}{60N}\left[\frac{C}{YF}\right]^3 \qquad (24)$$

where Y is the axial load factor supplied in accordance with AFBMA standards by Table V. If more than one bearing is mounted on a shaft, the rating life of the set is given by the following equation:

$$L_{10} = \left[\sum_{j=1}^{j=m} L_{10_j}^{-1.1}\right]^{-0.9} \qquad (25)$$

Example VI—Life of a triplex set

Compare the rating life of the duplex bearing set of Example IV with that of the triplex mounting of Example V. The basic load rating of each bearing is 5000 lb and the shaft speed 1000 rpm. The Y factor for a bearing having a 40-deg contact angle is 0.57 (Table V).

Duplex set:

$$L_{10_1} = \frac{10^6}{(60)(1000)}\left[\frac{5000}{(0.57)(1070)}\right]^3$$
$$= 9190 \text{ hr}$$
$$L_{10_2} = \frac{10^6}{(60)(1000)}\left[\frac{5000}{(0.57)(100)}\right]^3$$
$$= 11{,}250{,}000 \text{ hr}$$
$$L_{10} = [L_{10_1}{}^{-1.1} + L_{10_2}{}^{-1.1}]^{-0.9}$$
$$= [(9.19 \times 10^3)^{-1.1} + (11.25 \times 10^6)^{-1.1}]^{-0.9}$$
$$L_{10} = 9190 \text{ hr}$$

Triplex set:

$$L_{10_1} = \frac{10^6}{(60)(1000)}\left[\frac{5000}{(0.57)(590)}\right]^3$$
$$= 54{,}800 \text{ hr}$$
$$L_{10_2} = \frac{10^6}{(60)(1000)}\left[\frac{5000}{(0.57)(180)}\right]^3$$
$$= 1{,}929{,}000 \text{ hr}$$
$$L_{10} = [(2)(5.48 \times 10^4)^{-1.1} + (1.929 \times 10^6)^{-1.1}]^{-0.9}$$
$$= 54{,}100 \text{ hr}$$

Thus, the triplex set may have six times longer life than a duplex set. We have already seen that the triplex set yields less deflection under the same load—0.0002 in. as against 0.0003 in. for the duplex bearings. This is a significant gain in bearing design for machine tools, especially if the loads are larger than those chosen for this example. But the main drawback of a triplex set, of course, is that it requires more space, weight and cost.

PRELOADING TO PREVENT SKIDDING

Many modern applications such as aircraft gas turbines rotate at very high speed, 10,000 rpm and above, under light radial loading. This combination of light load and high speed tends to cause skidding, in lieu of rolling, between the bearing inner ring and rolling elements. Skidding, in turn, causes wear of the bearing load-carrying surfaces and results in decreased bearing life.

The cure for the skidding problem is higher specific ball or roller loads; ideally all rolling elements should be loaded. In radial ball bearing applications, this is easily accomplished by applying a light thrust load to the bearing. Also, a light uniform radial preload could eliminate skidding; however, it is mechanically easier to control a simple axial preloading device than it is to control a uniform radial preloading device, especially when the bearings are to operate under elevated temperatures.

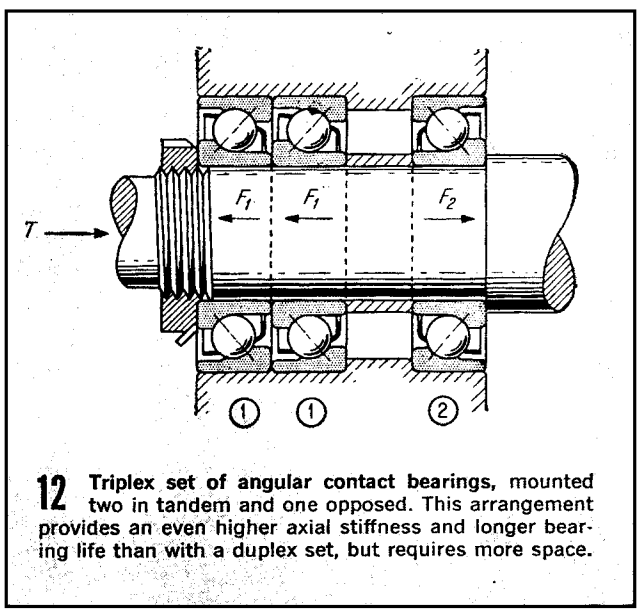
When an axial preload cannot be applied, as in cylindrical roller bearings, then one solution is the use of an out-of-round bearing outer ring wherein the major axis is aligned in the direction of radial loading. This has successfully prevented skidding. The roller load distribution becomes concentrated in two areas (Fig 13).

Generally, the amount of out-of-roundness is one order of magnitude larger than the bearing clearance, eg, 0.020 to 0.040 in. as compared to 0.001 in. clearance. To manufacture the out-of-round ring, a circular ring is distorted to the specified condition of out-of-roundness and ground circular. It is then allowed to spring back to its original condition. This process produces an inverted out-of-round ring. The outer ring is, of course, distorted to assemble over the rollers and cause a selective interference fit which is described by the equations below in terms of the roller angular location, ϕ:

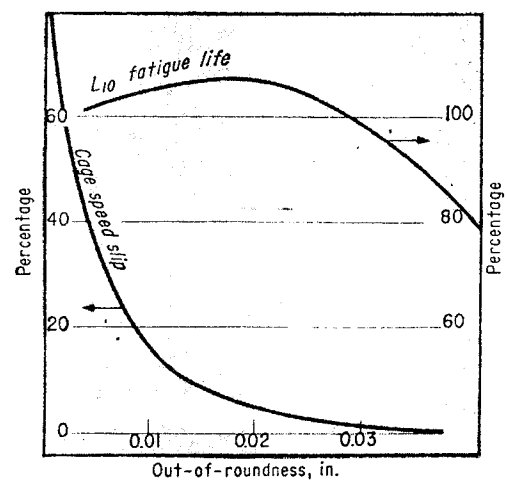
When $0 \leq \phi \leq \pm \frac{\pi}{2}$

$$C_k = R\frac{\left[\frac{2}{\pi} - \frac{1}{2}\cos\phi - \frac{1}{2}\phi\sin\phi\right]}{\frac{\pi}{2} - 1} + \frac{C}{2} \qquad (26)$$

When $\pm \frac{\pi}{2} \leqq \phi \leqq \pi$



12 **Triplex set of angular contact bearings,** mounted two in tandem and one opposed. This arrangement provides an even higher axial stiffness and longer bearing life than with a duplex set, but requires more space.



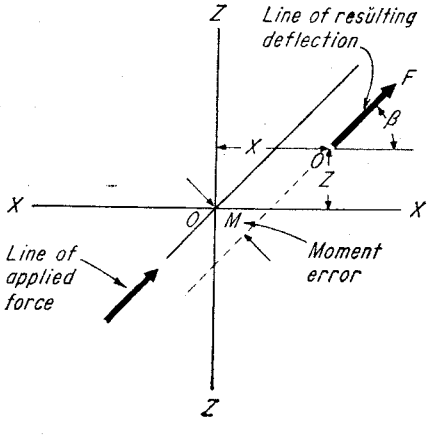
13 **Effect of out-of-roundness on slippage of cage and fatigue life.** Increasing the out-of-roundness decreases slip, but the fatigue life starts to drop off after a point as evident from the top curve.

$$C_k = R\frac{\left[\frac{2}{\pi} - \frac{1}{2}\cos(\pi-\phi)\right]}{\frac{\pi}{2}-1} - R\frac{\left[\frac{1}{2}(\pi-\phi)\sin(\pi-\phi)\right]}{\frac{\pi}{2}-1} + \frac{C}{2} \quad (27)$$

Where R = out-of-roundness, and C_k = clearance at angular location ϕ_k, (angle ϕ is equal to zero at the position of peak loading, Fig 3), and where the 0 to 180 deg axis is aligned in the radial load direction.

Because the out-of-round outer ring is supported in the housing at virtually two points, 0 and 180 deg, the outer ring is very flexible under roller load. This feature makes the bearing life less sensitive to out-of-roundness than it is to a uniform radial preload. Consequently, it is possible to avoid skidding and yet have satisfactory life.

As the degree of out-of-roundness is increased the cage-speed slip is reduced, (Fig 13), and the L_{10} fatigue life increases but then drops off due to over preloading. Analysis of skidding phenomena is difficult, but SKF has developed an IBM 7090 computer program which estimates the extent of skidding in high-speed roller bearings and further estimates the effectiveness of degrees of outer ring out-of-roundness in minimizing skidding. Bearing users may contract with SKF for the use of the computer program as it pertains to their application.



14 Effect of disturbing force F on the center of gravity of spring mass. It is frequently desirable to obtain isoelasticity in bearings in which the displacement in any direction is in line with the disturbing force.

PRELOADING FOR ISOELASTICITY

It is sometimes desirable that the axial and radial yield rates of the bearing and its supporting structures be as nearly identical as possible. In other words, a load in either the axial or radial direction should cause identical deflections (ideally). This necessity for *isoelasticity* in the ball bearings came with the development of the highly accurate, low drift inertial gyros for navigational systems, and for missile and space guidance systems. Such inertial gyros usually have a single degree of freedom tilt-axis and are extremely sensitive to error moments about this axis.

Consider a gyro in which the spin axis, Fig 14, is coincident with the X-axis, the tilt axis is perpendicular to the paper at the origin 0, and the center of gravity of the spin mass is located at the intersection of the three coordinate axes. If the spin mass is acted upon by a disturbing force F in the X-Z plane and directed at an oblique angle β to the X-axis, this force will tend to displace the spin mass center of gravity from 0 to $0'$. If, as shown by Fig 15, the displacements in the directions of the X and Z axes are not equal, the force F will create an error moment about the tilt axis.

In terms of the axial and radial yield rates of the bearings, the error moment, M, is

$$M = \tfrac{1}{2}F^2(R_z - R_x)\sin 2\beta \quad (28)$$

where the bearing yield rates R_z and R_x are in in./lb of force.

To minimize M and subsequent drift, R_z must be as nearly equal to R_x as possible — a requirement for pinpoint navigation or guidance. Also, from Fig 14 it can be noted that improving the rigidity of the bearing, ie, decreasing R_z and R_x collectively, reduces the magnitude of the minimal error moments achieved through isoelasticity.

What are the best ways for obtaining equal yield rates? In most radial ball bearings, the radial rate is usually smaller than axial rate. This is best overcome by increasing the bearing contact angle which reduces the axial yield rate and increases radial yield rate. You can achieve one-to-one ratios by specifying bearings with contact angles that are 30 deg or higher.

At these high angles, the sensitivity of the axial-to-radial yield-rate ratio to the amount of preload is quite small. It is, however, necessary to preload the bearings to maintain the desired contact angles.

Compact Ball Transfer Units

Masses of ball transfer units in airplane floor make it easy to shove cargo loads in any direction.

Compact ball transfer units roll loads every which way

An improved design of an oft-neglected device for moving loads—ball transfers—is opening up new applications in air cargo planes (photo above) and other materials handling jobs. It can serve in production lines to transfer sheets, tubes, bars, and parts.

Uses of established ball transfer units have been limited largely to furniture (in place of casters) and other prosaic duties. With new design that takes fuller advantage of their multiple-axis translation and instantaneous change of direction, ball transfer units can be realistically considered as another basic type of anti-friction bearing. The improved units are made by General Bearing Co., West Nyack, N.Y.

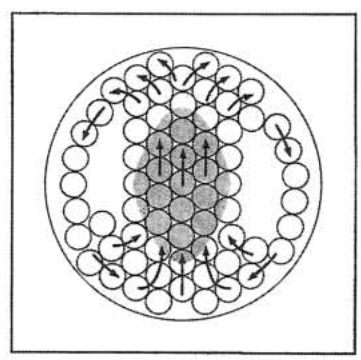
How they work. Essentially, ball transfers (photo below) are devices that translate omnidirectional linear motion into rolling motion to provide an unlimited number of axes of movement in any given plane. In such a unit, a large main ball rotates on its own center within a housing. This ball is supported by a circular group of smaller balls (drawing below) that roll under load and, in so doing, recirculate within the housing in endless chains.

These units are designed either as "ball up" or as "ball down." In the "ball down" units, design must provide a positive means of recirculating the support balls so they won't fall away under their own weight.

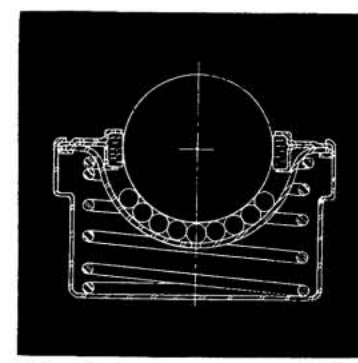
Variations. Many different configurations are available to suit the specific requirements of customers. Balls of carbon steel are most often used, but stainless steel balls are available for uses where corrosion may be a problem. Ball transfer units can be sealed to exclude dirt.

Where loads require that a number of ball transfers must simultaneously contact the load surface, a spring technique has been developed. Each ball transfer (drawing below) is spring-loaded. It starts to deflect when its own rated load is exceeded, allowing other ball transfers to pick up their share of the load. This concept also provides protection against major overloads in any ball transfer unit. □

Main ball shown, 1 in. dia., is supported by 70 smaller balls, hidden.



Shaded area seen from above shows load; arrows show ball circulation.



Spring loading assures even distribution of the load on the small balls.